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U. S. ARMORED CRUISER *NORTH CAROLINA.*

DESCRIPTION OF MACHINERY AND OFFICIAL TRIALS.

BY COMMANDER THEO. C. FENTON, U. S. N., RETIRED.

The armored cruiser *North Carolina* is one of two sister ships (the other being the *Montana*) built by the Newport News Shipbuilding and Drydock Company, of Newport News, Va. The contract for this vessel was signed January 3, 1905, the price being $3,575,000. This price does not include the armor and armor bolts (exclusive of protective deck), ordnance and ordnance outfit and certain articles supplied by the Government. The contract time for completion was thirty-six months. Owing to various delays for which the contractor was not responsible, this time was extended to April 27, 1908.

The main engines were required to develop twenty-three thousand indicated horsepower when making one hundred and twenty revolutions per minute, with a steam pressure of two hundred and fifty pounds at the high-pressure cylinder.

The guaranteed speed of the ship was twenty-two knots per hour for four hours.

PRINCIPAL DIMENSIONS OF HULL.

Length on load water line, feet	502
between perpendiculars, feet	502
over all, feet and inches	504–6
Breadth, molded, feet and inches	72–6
extreme to outside of plating, feet and inches	72–10½
Trial displacement, tons, about	14,500
draught to bottom of keel, feet	25
Tons displacement per inch immersion at normal draught	59.70
Bunker capacity to six inches below beams (43.5 cubic feet to the ton), tons	1,954
Capacity of engine-room feed tanks, tons	51
reserve feed-water compartments, tons	192

ELECTRIC PLANT.

There is installed and fitted complete an electric generating plant as follows:

Six 100-kilowatt generating sets, all of 125 volts pressure at the terminals. These sets are located in two independent dynamo rooms, known as the starboard dynamo room and the port dynamo room; three sets in each room.

The generating sets and the engines and dynamos conform in all respects to the latest requirements of the specifications for the United States Navy. Each generating set consists of an electric generator directly coupled to a steam engine, and both mounted on a common bedplate. They are required to run continuously for long periods under full load without variation in speed, with either vacuum or atmospheric exhaust, and worked at pressures twenty per cent. above or below the normal pressure of one hundred and fifty pounds, but only ninety per cent. of the full load need be carried while the engine is working twenty per cent. below the normal pressure on atmospheric exhaust.

ANCHOR WINDLASS.

The anchor windlass is of the vertical type and has two vertical shafts driven by worm gearing direct from a worm located on the crank shaft of the engine, without the intervention of counter shafts or beveled gears. Each shaft carries on its upper end, above the main deck, a wildcat or lock-

ing gear complete. The arrangement is such that the wildcats can be operated together or independently of each other. The wildcats revolve in a horizontal plane, taking in each bower chain on the inboard side and the sheet chain on the outboard side of the wildcats. The vertical shafts have couplings, and the lower end is keyed and supported by steady bearings.

The wildcat is cast from the best open-hearth steel to suit the 2¾-inch cable, as manufactured by the Government.

Each wildcat is fitted with a positive locking device worked by raised cams on the periphery of the locking rim and slotted keys operated by means of a lever. The entire operation of locking or unlocking the wildcat can be accomplished by one motion of the lever through an angle not exceeding sixty degrees.

A friction-band brake is fitted to each wildcat. The brakes have sufficient surface and ample strength in all parts to "ride by" with the windlass unlocked. Each end of the friction band is provided with a compressor, so that the bower or starboard chain can be checked when running out. The compressor is controlled by a screw hand wheel set close to the wildcat.

The windlass engine is designed for a working steam pressure of one hundred and fifty pounds per square inch, but is able to withstand the full boiler pressure.

The windlass was fully tested on the official trials of the vessel as follows:

The starboard anchor was let go in thirty-three fathoms of water and the chain veered to sixty fathoms; then the port anchor was let go and both chains veered until the starboard ninety-fathom shackle was at the hawse pipe and the port sixty-fathom shackle at the controller. The wildcats were connected, and both chains hove in simultaneously, at a uniform speed, until the port anchor was at the hawse pipe. The rate of heaving in both chains together was six fathoms in 42.6 seconds. The port anchor was at the hawse pipe in 6 minutes 52 seconds. The port wildcat was disconnected

and the starboard anchor was up in 3 minutes 46 seconds. The wildcats took the shackles and swivel without surging. There was no heating of thrust block or worm wheels.

STEERING ENGINE.

The steering gear is located aft in the compartment provided for it, and is of the standard type of the Bureau of Construction and Repair, consisting of a right- and left-hand screw with traversing nuts directly connected, by side rods, to a crosshead on the rudder stock. The crosshead is secured to the rudder stock by keys. The weight of the rudder is transmitted to this crosshead by means of a wrought-steel ring fitted in a rabbet near the head of the rudder stock, between the crosshead and the casting on the stern post. A floating disc of phosphor bronze is fitted and recessed in the stuffing-box casting to hold oil, the weight of the rudder being taken by this floating ring. The casting is bolted to the stern post, taking the weight of the rudder through this floating ring, and is fitted with a stuffing box around the rudder stock capable of adjustment to convenient location in the steering space. A friction band is fitted to the rudder stock to be operated from the engine steering room.

The steering engine is of compact and efficient design, and is capable of putting the rudder hard over in twenty seconds, when the vessel is moving ahead at full speed, at a working steam pressure of one hundred and fifty pounds per square inch. The engine is of sufficient strength to withstand the operation under full boiler pressure.

Provision is made in the steering-engine room for hand steering on a shaft geared to the main screw shaft. A slip joint is provided between the engine and screw shaft to take up the lateral motion.

Direct hand steering is also provided for in an emergency, by suitable arrangement of relieving tackles, directly connected to the crosshead on the rudder stock. The hand-steering wheels are arranged to be thrown out of gear when the engine is at work.

The steering gear was fully tested on the official trials, and found entirely satisfactory.

PROPELLING MACHINERY.

The propelling engines are right and left, and are placed in watertight compartments, separated by a middle-line bulkhead.

The engines are of the vertical, inverted cylinder, direct-acting, triple-expansion type. The order of the cylinders, beginning forward, is forward low pressure, high pressure, intermediate pressure and after low pressure. The forward low-pressure and high-pressure cranks are opposite, also the intermediate and after low-pressure cranks, the second pair being at right angles with the first. The sequence of cranks is, therefore, high pressure, intermediate pressure, forward low pressure and after low pressure.

The main valves are worked by the Stevenson link motion with double-bar links. There is one piston valve for the high-pressure and two each for the intermediate and low-pressure cylinders.

The framing of the engines consists of forged columns trussed by forged-steel stays. The engine bedplates are of cast steel supported on the keelson plates. The crank, line and propeller shafting is hollow. The shafts, piston rods, connecting rods and working parts generally are of forged steel.

REVERSING GEAR.

The reversing gear for each engine consists of a steam cylinder, and an oil-controlling cylinder bolted to the high-pressure cylinder. The common piston rod of the reversing-gear cylinders acts directly on arms keyed to the reversing shaft. The piston rod passes through the controlling cylinder with a uniform diameter.

The valve of the steam cylinder is of the piston pattern, of composition, working in a composition-lined valve chest.

There is a by-pass valve on the oil cylinder, worked by a

continuation of the stem of the steam-piston valve. These valves are worked by a floating lever, the primary motion being derived from the hand lever on the working platform and the second motion from the reversing shaft, all parts being so adjusted that the reversing shaft follows the motion of the hand lever and is firmly held when stopped. There is a stopcock in the by-pass pipe of the oil cylinder, and a pump for reversing by hand is connected to the oil cylinder with its lever convenient to the working platform. The by-pass pipes are connected to the valve box of the hand pump in such a way as to leave the hand arrangements always in gear. The piston of the oil cylinder is packed by two cup leathers.

REVERSING SHAFT.

There is one reversing shaft for each engine, with an axial hole through it. It has arms for the reversing engine and for each link. Each reversing arm for the links is made with a slot fitted with a block, to which the extension links are attached. Each block is adjustable in the slot of its arm by a screw and hand wheel of an approved hand-locking device and is fitted with a suitable index. The slots in these arms are so arranged that the links may be thrown into full backward gear irrespective of the position of the block in the slot; and the length of the slots is such that cutoff may be varied from about $\frac{5}{100}$ to $\frac{75}{100}$ of the stroke.

TURNING ENGINES AND GEAR.

There is installed in each engine room a double engine for turning the main engine with steam of one hundred pounds pressure. This engine drives, by worm gearing, a second worm, which may be made at will to mesh with a worm wheel fitted on the crank shaft.

The turning engines have piston valves, and are made reversible by means of Stevenson links, reversed by hand levers.

Each turning engine shaft is fitted for turning by hand.

PROPELLERS.

There are two 3-blade propellers, both outboard turning, for ahead motion. The blades and hub are of manganese-bronze. The bolt holes in the flanges of the blades are made oval to allow of adjustment of the pitch, and each blade is firmly bolted to the hub and secured from turning by lock plates.

The dimensions of the propellers are as follows:

Diameter, feet	18
Pitch, as set, mean, feet and inches	22–06
adjustable from 21 feet 6 inches to, feet and inches	23–06
Ratio of diameter to pitch	.8
Area, projected, square feet	78
helicoidal, square feet	100
disk, square feet	254.47
Height of lower tip of blade above keel, inches	16.7
Immersion of upper tip of blade at low draught, inches	67.3

After the official trials the pitch was set to 21 feet 9 inches.

ENGINE DATA.

Cylinders, number for each engine	4
H.P., diameter, inches	38½
I.P., diameter, inches	63½
F.L.P., diameter, inches	74
A.L.P., diameter, inches	74
Stroke of all pistons, inches	48
Valves, H.P., one for each cylinder, diameter, inches	23
I.P., two for each cylinder, inches	25½
L.P., two for each cylinder, inches	25½
travel of, inches	10½
Piston rods, diameter, inches	8½
axial holes, H.P. and I.P., inches	4½
L.P., inches	5
Connecting rod, length from center to center, inches	96
crosshead end, diameter, inches	8½
crank end, diameter, inches	9½
Crank shaft, number of sections	2
diameter, inches	18½
axial hole, inches	10
length, feet and inches	35–04
Coupling disks, diameter, inches	32
thickness, inches	4½
Crank pin, diameter, inches	20
axial hole, inches	12
length, feet and inches	1–11

Thrust shaft, diameter, inches	17½
axial hole, inches	10½
length, feet and inches	25-11½
Stern-tube shaft, diameter, inches	18½
axial hole, inches	10½
length, feet and inches	52-00½
Propeller shaft, diameter, inches	18½
axial hole, inches	10½
length, feet and inches	48-05½

Thrust shaft and line shaft combined.

CONDENSERS.

Main Condenser.—There is one main condenser in each engine room, oval in form, the inside dimensions being 9 feet 4 inches high, 5 feet 8 inches wide, the top and bottom being struck with a radius of 2 feet 10 inches. The shell is $\frac{5}{16}$-inch thick with two double butt joints, and with circumferential and longitudinal angle and T-bar stiffeners.

The water chests are cast $\frac{5}{8}$-inch thick. The forward chest, being the one for the entrance and exit of circulating water, has a horizontal division plate in the middle fitted with valves which, when open, allow the circulating water to pass overboard direct, the valves being worked by a lever on the outside of the condenser, and it has nozzles for inlet and outlet of circulating water twenty-one inches in diameter.

The condenser-tube sheets are rolled, each in one piece. They are one inch thick, with smoothly finished holes for the tubes, tapped and fitted with screw glands for packing the tubes.

There are 6,292 seamless-drawn tubes in each condenser, $\frac{5}{8}$-inch outside diameter, No. 16 B. W. G. in thickness. The tubes are 14 feet and $\frac{1}{2}$ inch long between tube sheets and are spaced $1\frac{5}{16}$-inch between centers. They are supported at two intermediate points by ferrules, $\frac{3}{4}$-inch long, inserted in supporting plates. Baffle plates are fitted to direct the steam over all the tubes.

The cooling surface for each condenser is 14,411 square feet, measured on the outside of the tubes.

The condensers are supported by angle plates riveted between circumferential angle bar stiffeners.

There is riveted to the shell of each condenser a casting with two composition nozzles with faced flanges twenty-seven inches diameter of opening, for attachment of the main exhaust pipes, and a faced flange nine inches in diameter of opening for the auxiliary exhaust pipe.

Cast-steel flanges are riveted to the shell of each condenser and properly faced for an air-pump suction pipe thirteen inches in diameter and for two manholes at the bottom twelve inches in diameter. There is a one-inch connection in the bottom manhole for cleaning the tubes by boiling.

Drain cocks are provided with pipes leading to the bilge.

A copper tank is provided for admitting an alkaline solution into each condenser, the tank having a capacity of five gallons. Zinc protectors and safety valves are fitted.

The material of the condenser is as follows: Shells, steel, class B boiler plate. Baffle plates, steel, class C boiler plates. Tubes, composition: copper 70, tin 1, zinc 29 per cent. Tube sheets and supporting plates are as near as possible the same material as the tubes. Glands are of tubing, and of the same composition as the tubes. Water chests are of composition. All bolts are of bronze.

AUXILIARY AND DYNAMO CONDENSERS.

The material of these condensers is as follows: Shells, steel, class B, boiler plate; tubes, composition, copper 70, tin 1, zinc 29 per cent.; tube sheets, as near as possible of the same material as the tubes; heads, composition.

The cooling surface of the auxiliary condensers is 600 square feet, and of the dynamo condensers 700 square feet, measured on the outside of the tubes.

In each engine room there is an auxiliary condenser connected from the auxiliary exhaust pipe to all the auxiliary machinery. Each condenser has an air and a circulating pump.

In each dynamo room there is an auxiliary condenser for

the exclusive use of the dynamo engines. Each has an air and a circulating pump. The tube sheets are one inch thick. The diameter and the spacing of the tubes and the packing are the same as in the main condensers.

The tubes of the auxiliary and the dynamo condensers are of the same length.

EVAPORATING AND DISTILLING PLANT.

The evaporating and distilling plant is placed on the protective deck. There are four evaporators and four distillers with their accessories.

The evaporators have a combined capacity of 23,000 gallons of water per twenty-four hours.

The distillers have a combined capacity of 23,000 gallons of potable water per twenty-four hours.

The evaporators take steam from the auxiliary steam pipe, and the coil drain pipes lead through a by-pass, automatic traps, to the feed tanks. The evaporator feed- and fresh-water pumps take steam from the evaporator coils, as well as from the auxiliary steam pips.

The shells of the evaporators have connections with valves and pipes for directing the steam into the distillers and into the auxiliary exhaust pipe.

The feed water for the evaporators is taken from the circulating pipe, after it has passed through the distillers, and from the sea.

There are blow pipes of ample size, so arranged that the evaporators may be blown out when working under pressure.

The distiller circulating pumps discharge to the distillers, the water passing overboard or into the sanitary pipe at will. There is also a direct connection from this pump to the sanitary pipe, and the relief valve for the distillers is so placed as to act for both the distillers and the sanitary pipe.

In addition to the pump connection for distiller circulating, provision is made for circulating water through the distillers, from the most conveniently located discharge pipe from the pump to the fire main.

A small reservoir tank, fitted with a removable cover, is placed in the fresh-water pump suction between the pump and the distillers.

A water meter, without lead, is placed in the pipe connecting the reservoir tank with the bottom of the distillers.

Evaporators.—There are four horizontal evaporators, each having 320 square feet of tube-heating surface.

The tubes are two inches outside diameter and are without bends.

The tubes are secured to the tube sheets so that adequate provision is made for expansion, and the tubes are so arranged that their removal will leave the shell accessible in all its parts for scaling.

All tubes are easily removable for cleaning and repairs, and there are no internal detachable joints in the tubes.

Distillers.—There are four distillers, each having 115 square feet of tube-cooling surface, measured on the outside of the tubes.

The tubes are straight, $\frac{5}{8}$-inch outside diameter, tinned on both sides, and well expanded and sweated into the tube sheets.

Provision is made for the expansion of the tubes by the use of a flanged tube sheet working a stuffing box.

FEED-WATER HEATERS.

In each engine room there is a feed-water heater with all necessary fittings complete.

The heating surface of each heater is 1,150 square feet, measured on the outside of the tubes. They are of the direct-flow type, located on the discharge side of the main feed pumps.

The heating agent is the auxiliary exhaust steam.

The tubes are $\frac{5}{8}$-inch outside diameter, No. 16 B. W. G.

PUMPS.

The pumps are in accordance with the following table:

Auxiliary.	Number.	Type, make and location.	Steam cylinders.					Water cylinders.			
			Number.	Diameter, inches.	Diameter piston rod, inches.	Stroke, inches.	Constant "K" (mean of both ends).	Number.	Diameter, inches.	Diameter piston rod, inches.	Stroke, inches.
Main air pumps.	2	Cameron, twin, vertical bucket, single-acting, one in each engine room.	2	14	2¾	18	.00686	2	35	3¾	18
Main circ. pumps.	2	Centrifugal, driven by compound engine, one in each engine room.	2	H.P., 11 L.P., 22	2½	10	.00235 .00955	...	...	...	...
Main feed pumps.	4	Cameron, vertical piston, double-acting, single, two in each engine room.	1	16	2¾	18	.009	1	10	3	18
Aux. feed pumps.	4	Cameron, vertical piston, double-acting, single, one in each starboard fireroom.	1	12	2½	12	.00337	1	8	2½	12
Reserve feed-water pump.	1	Cameron, vertical piston, double-acting, single, starboard engine room.	1	5	1	6	.00029	1	6	1	6
Fire and bilge pumps.	6	Cameron, vertical piston, double-acting, single, one in each engine room.	1	12	2½	12	.00337	1	10	2½	12
		One in each port fireroom.	1	10	1¾	12	.00234	1	8	2	12
Auxiliary condenser pumps.	2	Cameron, combined horizontal piston, single, one in each engine room.	1	6	1¼	12	.00084	1	10	1¼	12
Dynamo condenser pumps.	2	Cameron, combined horizontal piston, single, one in each engine room.	1	6	1¼	12	.00084	1	10	1¼	12
Evaporat. and distill. plant.	2	Evaporating feed pump, Cameron, vertical piston, double-acting, single, evaporating rooms.	1	3½	¾	4	.00028	1	4	¾	4
	2	Distilling fresh-water pumps, Cameron, vertical piston, double-acting, single, evaporating rooms.	1	3½	¾	4	.00028	1	4	¾	4
	1	Distilling circulating pump, Cameron, horizontal piston, double-acting, single, starboard evaporating room.	1	12	2½	12	.00337	1	14	2¾	12
Ice machine.	1	Distilling circulating pump, centrifugal, driven by De Laval st. turbine, port evaporating room.	...	...	...	...	...	...	...	...	...
	2	Allen dense-air, H. B. Roelker, berth deck, starboard, forward.	1	9¾	1¼	13	.00242	...	...	...	...

BOILERS.

There are sixteen Babcock & Wilcox boilers of the latest improved type, placed in eight watertight compartments.

The steaming capacity is such that all steam machinery on

board can be run at full power with an average air pressure in the firerooms of not more than two inches of water. The following list gives the particulars of these boilers:

Length, feet and inches	9–01½
Width, external, feet and inches	16–00½
Number of furnaces	2
headers	27
Grate surface, one boiler, square feet	99.37
Heating surface, one boiler, square feet	4,250
Grates, length, feet and inches	7–00
width, feet and inches	6.83
Pressure, design, working pounds	265
test, pounds	400
Ratio, G. S. to H. S	42.76
Number of tubes, one boiler, 2-inch	901
4-inch	30
Length of tubes, as fitted, feet and inches	8–02

REFRIGERATING PLANT.

There are two Allen dense-air ice machines, each capable of producing the cooling effect of two tons of ice per day. The cooling pipes from the machines are led into the ice tank, the scuttlebutts and the cold-storage room.

Valves are provided in accordance with the Bureau of Steam Engineering standard arrangement of valves, in the cold-air pipes of refrigerating plant so that the air may go to the cold-storage room direct, or through the ice-making tank, and thence to the cold-storage room and scuttle butts; and also from the ice-making tank direct to the scuttlebutts.

AIR COMPRESSORS.

There are three Westinghouse air compressors in the after part of the port engine room for use in running pneumatic tools in the Steam Engineering department, for blowing soot off the boiler tubes, and for the gas-ejecting system for the guns.

Each compressor has a capacity of about fifty cubic feet of free air per minute, at one hundred pounds pressure. The compressor is driven by steam.

OFFICIAL TRIALS.

Standardization of Screws.—The vessel was tried by the standardized-screw method, progressive runs being made over the measured mile off Rockland, Maine, on January 6, 1908. From the data obtained on these runs the curves shown on Plate 1 were plotted.

It was determined that 118.64 revolutions per minute would be required to make the true contract speed of 22 knots.

The draught and corresponding displacement at the beginning and the end of the runs were as follows:

	Beginning.	End.
Draught, forward, feet and inches	24–07¾	24–03¾
aft, feet and inches	25–10½	25–07
Displacement, tons	14,669	14,470

OFFICIAL FOUR-HOURS' TRIAL.

On January 7, 1908, the four-hours' official trial, prescribed by the contract, was held, the ship steering S.W. by S. The sea was very moderate. The draught and displacement at the beginning of this trial were as follows:

Forward, feet and inches	24–09¼
Aft, feet and inches	25–06¼
Corresponding displacement, tons	14,613

Draught and displacement at end of trial were as follows:

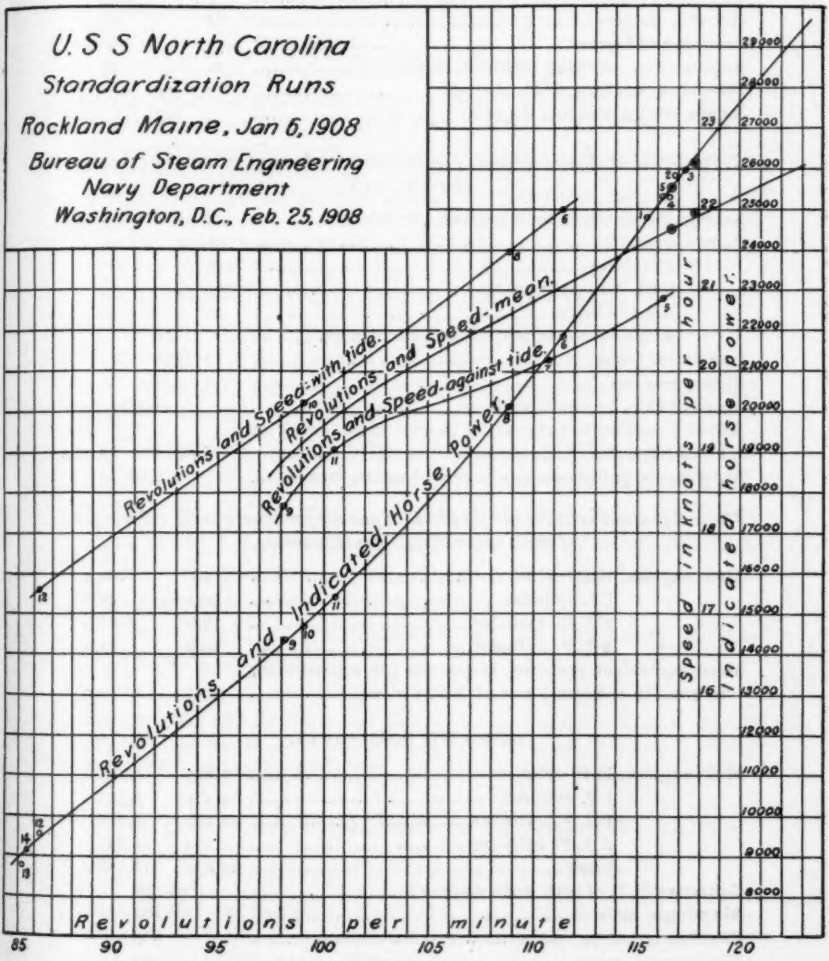
Forward, feet and inches	24–04¾
Aft, feet and inches	25–06
Corresponding displacement, tons	14,468

The data obtained on this trial were as follows:

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	271.8	
engines, pounds	257	258
H.P. steam chest, gauge, pounds	244	242
1st receiver (absolute), pounds	106	115
2d receiver (absolute), pounds	37	45
Vacuum in condensers, inches of mercury, mean	25.6	24.7


U. S S North Carolina
Standardization Runs
Rockland Maine, Jan 6, 1908
Bureau of Steam Engineering
Navy Department
Washington, D.C., Feb. 25, 1908
Revolutions and Speed-with tide.
Revolutions and Speed-mean.
Revolutions and Speed-against tide.
Revolutions and Indicated Horse Power.
Revolutions per minute
Speed in knots per hour
Indicated horse power
85
90
95
100
105
110
115
120
16
17
18
19
20
21
22
23
24
25
26
27
28
29
8000
9000
10000
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17000
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19000
20000
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22000
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24000
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26000
27000
28000
29000

Plate I.

Temperatures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Injection, degrees	42	42
Discharge, degrees	107	115
Hotwell, degrees	82	98
Feed water, degrees	173	170
Engine room, working platform, degrees	70	74
Firerooms, working level, degrees	77	
Smoke stacks, average, degrees	605	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per minute	118.29	117.62
Pumps, main air	45	45
circulating	210	221
feed, d.s., per minute	31	33
fire and bilge	26	29
auxiliary condenser, air and circulating	48	
Dynamo engines	350	
Blower engines	521.4	
Speed of ship, in knots per hour	21.918	
Slip of propeller, in per cent. of its own speed, based on mean pitch	16.56	16.06
Air pressure in firerooms, in inches of water, mean	2.28	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	112.8	100.5
I.P. cylinder	50.2	44.8
F.L.P. cylinder	18.6	22.3
A.L.P. cylinder	19.8	23.2
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	52.5	53.9

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	3,675	3,253
I.P. cylinder	4,484	4,373
F.L.P. cylinder	2,287	2,710
A.L.P. cylinder	2,424	2,832
total	12,870	13,168
Collective H.P. of both main engines	26,038	
Air pumps, main	560	
Circulating pumps, main	560	
Feed pumps, main	560	
Hot well	560	
Other auxiliaries	576	
Collective, main and auxiliary engines in operation	27,274	

COAL.

Kind and quality used on trial	New River, hand picked.
Pounds, per hour, main and auxiliary engines, during trial	71,548

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	17.15
Pounds of coal per I.H.P. per hour, main engines	2.748
All machinery in operation	2.623
Square foot of grate surface, per hour	44.96
Cooling surface (main condenser), square feet per I.H.P. main engines	1.0745
Heating surface, square feet per I.H.P. (total)	2.493

The speed of the ship on this trial was 21.914, which was a trifle below the contract requirement. The Department directed that a four-hours' full-power trial be held on February 15, 1908, with the view of obtaining the full contract speed.

This trial was held on the date specified. The weather was generally cloudy, with moderate to fresh breezes from N.W. to W. At the beginning of the trial there was a moderate swell from S.E., increasing to a long, heavy swell at 10·30 P. M., and remaining about the same to the end of the trial. The course was about N.E. magnetic.

The draught and displacement at the beginning of this trial were as follows:

Forward, feet and inches	24–11⅜
Aft, feet and inches	26–00½
Corresponding displacement, tons	14,847

The draught and displacement at the end of the trial were as follows:

Forward, feet and inches	24-04½
Aft, feet and inches	25-07⅜
Corresponding displacement, tons	14,430

The data obtained on this trial were as follows:

	Starboard.	*Port.*
Average revolutions	123.24	122.12
steam pressure at boilers	291	
H.P. steam chest	253	253
pressure in fireroom, inches water	1.81	
vacuum	25	25
I.H.P., main engines	29,225	
Speed of ship, in knots per hour	22.481	
Slip of propellers, in per cent. of their own speed	17.85	17.09

OFFICIAL TWENTY-FOUR HOURS' ENDURANCE TRIAL.

At 6·00 A. M., January 9, 1908, the twenty-four hours' endurance trial, under all boilers, as prescribed by the contract, was commenced. The weather was partly overcast at the beginning, with light westerly breezes, and a very moderate sea from S.E. During the afternoon and up to 1·00 A. M. the wind increased to a fresh gale, veering and hauling from N.W. to W.S.W. The weather moderated after 1·00 A. M. until clear, with light northerly airs at finish, with a smooth sea.

The data obtained on this trial were as follows:

PERFORMANCE.—TWENTY-FOUR HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	243.2	
engines, pounds	232.4	231.6
H.P. steam chest, gauge, pounds	216.8	216.5
1st receiver (absolute), pounds	85.0	96.0
2d receiver (absolute), pounds	27.7	34.0
Vacuum in condensers, inches of mercury, mean	25.7	25.1

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	47	48
Discharge, degrees	113	116
Hotwell, degrees	91	97
Feed water, degrees	183	164
Engine room, working platform, degrees	60	61
Firerooms, working level, degrees	73	73
Smoke stacks, average, degrees	561	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per minute	108.74	107.6
Pumps, main air	30	35
circulating	177	184

	Starboard.	*Port.*
Pumps, feed, d.s. per minute	19	24
fire and bilge	30	41
auxiliary condenser, air and circulating	53	42
Blower engines	420	
Speed of ship, in knots per hour	20.556	
Slip of propeller, in per cent. of its own speed, based on mean pitch	14.86	13.96
Air pressure in firerooms, in inches of water, mean	1.2	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	109.3	108.4
I.P. cylinder	43.2	43.6
F.L.P. cylinder	13.4	14.7
A.L.P. cylinder	15.2	16.1
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	44.7	46.1

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	2,216	3,119
I.P. cylinder	3,486	3,448
F.L.P. cylinder	1,490	1,596
A.L.P. cylinder	1,690	1,757
total	9,882	9,920
Collective H.P. of both main engines	19,802	
Air pumps, main	420	
Circulating pumps, main	420	
Feed pumps, main	420	
Hotwell	420	
Other auxiliaries	443	
Collective, main and auxiliary engines in operation	20,665	

COAL.

Kind and quality used on trial	New River; poor.
Pounds per hour, main and auxiliary engines, during trial	39,135

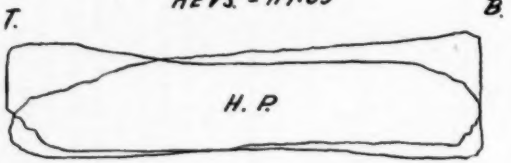
DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	13.42
Pounds of coal per I.H.P. per hour, main engines	1.98
all machinery in operation	1.87
square foot of grate surface, per hour	24.6
Cooling surface (main condenser), square feet per I.H.P., main engines	1,366
Heating surface, square feet per I.H.P. (total)	3.19

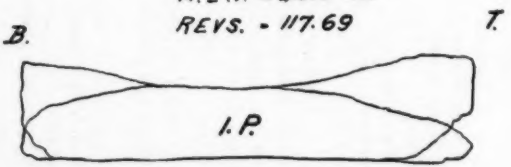
"NORTH CAROLINA."—INDICATOR CARDS TAKEN ON FOUR-HOUR TRIAL, JANUARY 7, 1908.

Starboard engine.

M.E.P. = 115 lbs.
REVS. = 117.69

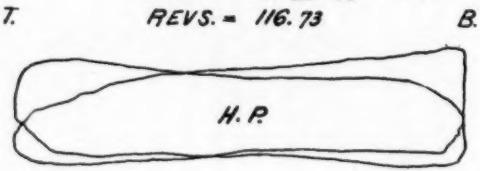


M.E.P. = 46.5 lbs.
REVS. = 117.69

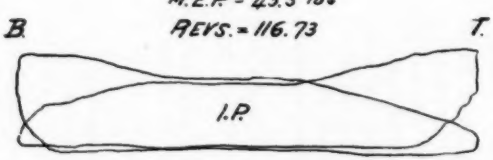


Port engine.

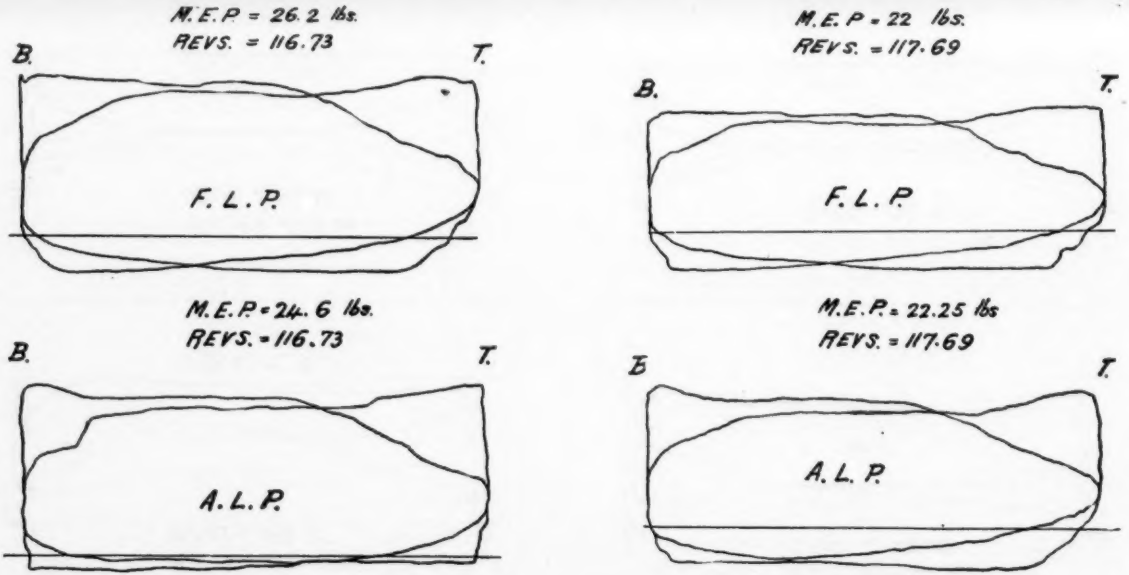
M.E.P. = 101.25 lbs.
REVS. = 116.73



M.E.P. = 43.5 lbs
REVS. = 116.73

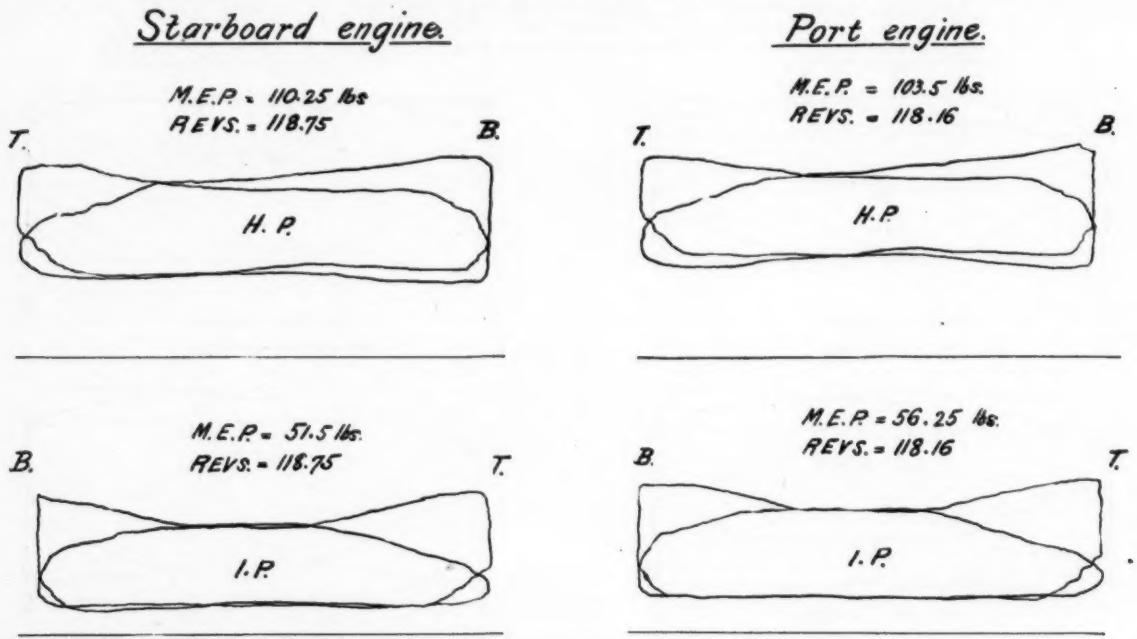


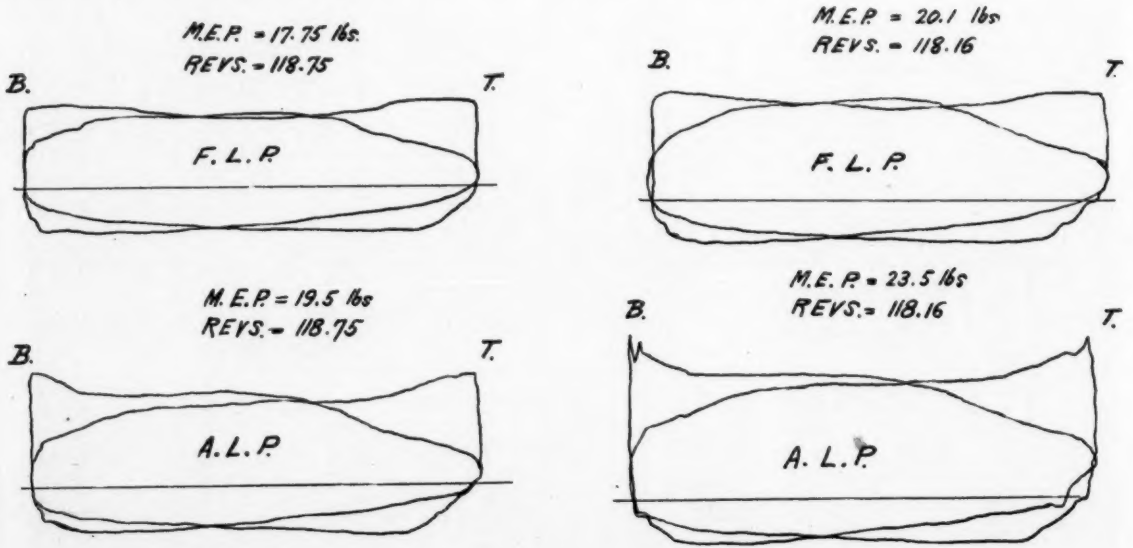
M.E.P. = 26.2 lbs.



Starboard.		*Port.*
265	Pressure at engine, gauge	265
250	H.P. chest, gauge	240
107	I.P. receiver, absolute	125.7
42	L. P. receiver, absolute	50.7
25.4	Vacuum, inches	24.3

"NORTH CAROLINA."—INDICATOR CARDS TAKEN ON TWENTY-FOUR HOUR TRIAL, JANUARY 9, 1908.





Starboard.		*Port.*
180	Pressure at engine, gauge	165
175	H.P. chest, gauge	157
77	I.P. receiver, absolute	70
26	L.P. receiver, absolute	23
26.4	Vacuum, inches	25.3

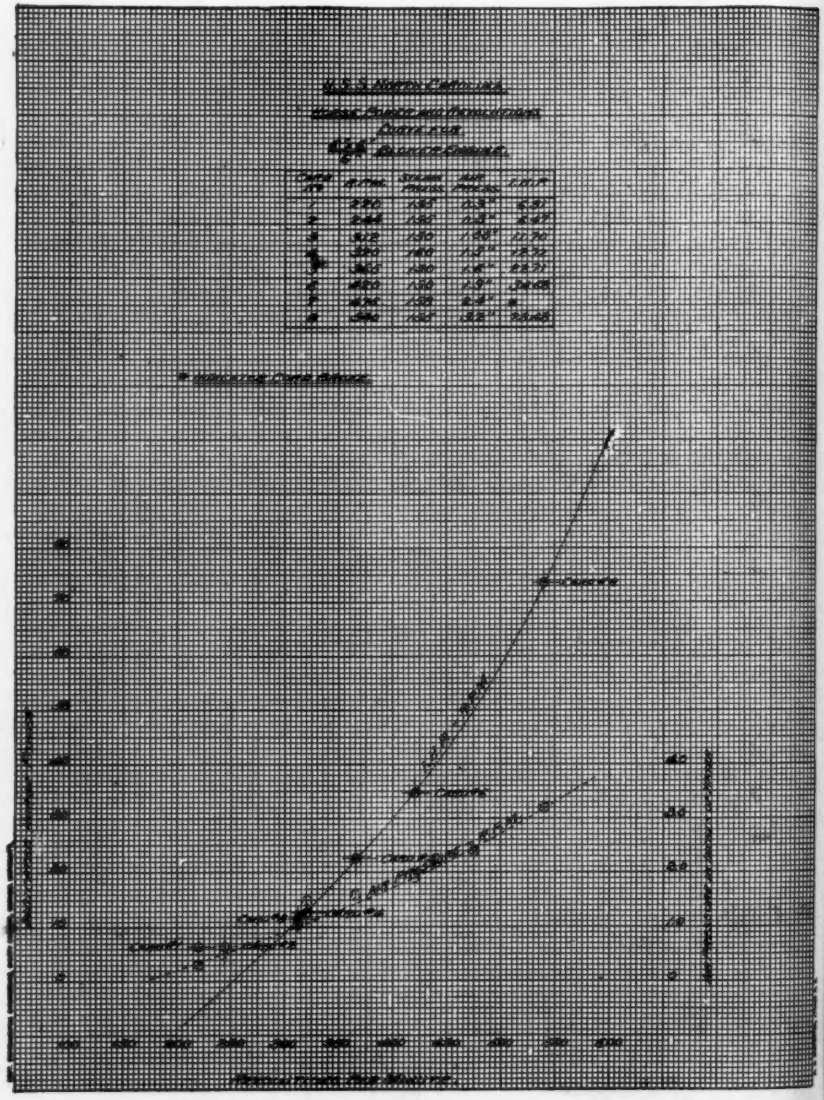


Plate II.—"North Carolina." Curve for Blower Engine.

THEORY OF THE SCREW PROPELLER.

By von J. W. Haeussler.

Translated by A. M. P. Maschmeyer, Associate.

INTRODUCTION.

The adaptation of the steam turbine for marine purposes has given an increasing importance to the design of the propeller, and a need for theoretical formulae for the calculation of the same has been felt. For vessels having reciprocating engines empirical formulae have been deduced for the design of the screw. When the greater number of revolutions of the steam turbine had to be transferred to the propeller these formulae proved useless. For want of knowledge of the theory of fast-turning propellers trials were relied upon; for each new vessel several screws were made, and by trial trips the one giving the highest efficiency was chosen. In the present article the design of the screw will be deduced by mechanics. Efforts in this direction have, to my knowledge, not been published; I have been unable to find anything under kindred subjects that could be compared with my theoretical representations.

The theories of the actions of the screw propellers known up to the present time are deduced from dynamics.

Dynamics and mechanics are correlated conceptions, the former treating of the theory of forces and the latter of the theory of the motions produced by acting forces.

If we consider the action of a propeller as a translation of its torque into a rectilinear work of the vessel, then it is unnecessary to know the magnitude of the hydraulic work;

it will be sufficient if we are able to ascertain in per cent. the loss of work which must be deducted from the work of the screw as loss of the thrust.

The work of the screw is composed as the hydraulic pressure of two components—the axial and tangential components. The former produces the useful work of propelling the vessel, while the latter is the loss in moving the water. The latter again is dependent upon the construction of the screw, as we will see, and can be expressed and subtracted in fractions of the propeller efficiency.

The friction of the screw in the water is also a loss which in a similar manner can be expressed and eliminated by an economical coefficient.

The present article is not intended to prove theoretically the efficiency of any oftentimes unsuitably-designed screw; such generalization would only confuse and make the contents more difficult to understand without serving a useful purpose.

The theory must show the way how to design a screw so as not to conflict with the laws of mechanics, and what dimensions and angles should be adapted for each case.

The mechanical theory of the hydraulic screw as presented by me is closely related to the theory of the aerial screw propeller, and professional circles labor to make it stand for the study of power flight. (*Zeitscher d. V. D. T.*, 1906, No. 43, p. 1766.) To answer both purposes and to be able to use the obtained equations equally well for liquids as for gases has been my object, and therefore the theories have been given in general.

With some modifications the herewith-given theory of the screw propeller can be applied to flying machines, in which two opposed-bladed aerial screw propellers are arranged above each other, and turning in a horizontal plane against each other, so that the slip of the basket or frame which would be caused by the employment of only one screw would be counteracted by the slip of the second screw.

The theory treats first of the reciprocal actions between a

moving fluid and a plane with consideration of the pitch or angle used. After that the obtained equations will be introduced in the mechanic.

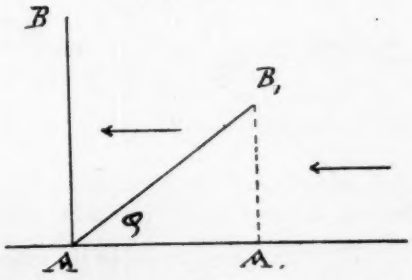
§1.—Pressure of a Moving Fluid Against a Quiescent Plane and of a Moving Plane Against a Quiescent Fluid.

If a fluid (liquid or gas) moves against a plane of length l and breadth b, and p is the normal pressure of the fluid upon the unit surface of the plane, then the total pressure P_1 normal to the plane is

$$P_1 = lbp. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In Fig. 1 line AB represents the breadth b of the supposed plane and the arrows indicate the direction of motion of the

Fig. 1



fluid normal to the plane $(AB)l$. The molecules act against the plane with a pressure equal to the amount expressed by equation (1).

We can also imagine that the plane is oblique to the direction of motion of the fluid, so that its breadth in Fig. 1 can be represented by line AB_1. In this oblique position only as many molecules act against the plane as will pass through the rectangle $(A_1B_1)l$. The total pressure P_2 against the oblique plane is therefore $P_2 = (A_1B_1)lp$.

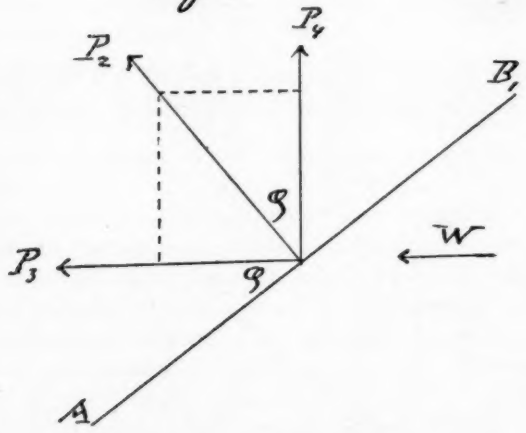
If φ is the angle of the oblique plane with the direction of motion of the fluid, then $A_1B_1=AB_1 \sin\varphi=b \sin\varphi$.

Substituting this value in the previous equation, the total pressure then becomes

$$P_2=lbp \sin\varphi. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The direction of this pressure is normal to the plane, but the impelling molecules of the fluid continue their motion from the plane through an angle of the same magnitude.

Fig 2



In Fig. 2 the line AB_1 represents again the cross section of the plane, and the arrow W the direction of motion of the fluid; and if plane AB_1 is movable it will be forced in the direction of P_2 which is normal to the plane $(AB_1)l$. The pressure P_2 can be divided into two components normal to each other. The component P_3 in the direction of the moving fluid is

$$P_3=P_2 \sin\varphi, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

while the component P_4, opposite to the direction of the fluid coming from the plane, is

$$P_4 = P_2 \cos\varphi. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This can also be deduced from the parallelogram of forces of which P is the diagonal; for

$$P = \sqrt{P_2^2 \sin^2\varphi + P_2^2 \cos^2\varphi + 2P_2 \sin\varphi \, P_2 \cos\varphi \cos 90^\circ}.$$

But

$$\cos 90^\circ = 0,$$
$$\sin^2\varphi + \cos^2\varphi = 1;$$

therefore

$$P = P_2; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

or substituting for P_2 its value from equation (2)

$$P = lbp \sin\varphi. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The component in the direction of motion of the fluid results from the equations (2) and (3)

$$P_3 = lbp \sin^2\varphi, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

and the component P_4 normal to the direction of motion of the fluid from equations (2) and (4)

$$P_4 = lbp \sin\varphi \cos\varphi. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

In the above considerations it is supposed that the plane is displaced only relatively to its breadth by the direction of motion of the fluid. But a displacement relatively to its length can also be considered. In such case the entering angle φ changes, but a new angle does not enter into the given equations. We can compare the direction of motion of a fluid against a plane with a ray of light that is reflected from the surface of a mirror. As is well known, here only one angle is to be considered, which can be compared with our entering angle φ.

All the given equations are applicable not only to a moving fluid against a quiescent plane, but also vice versa, viz: to a moving plane against a quiescent fluid. The relative change

in position is in both cases the same, as also the resulting pressure. The angle φ is determined by the position of the plane relatively to its direction of motion.

§2. Determination of Pressure p.

The common equation for moving bodies

$$ps=\frac{1}{2}mv^2, \quad \ldots\ldots\ldots \quad (9)$$

which gives the change of kinetic energy into work, cannot be directly employed in the present case for the determination of the specific surface pressure p, but has to be transformed for a constantly recurring work.

Let us consider a fluid of the mass m which constantly moves with a velocity v against a plane.

The molecules repelled by the plane are replaced in the above case constantly by new ones; therefore, if the plane is movable it will be propelled so that the moving molecules exert a constant force upon the plane.

The path s in the above motion equation at the beginning of the work is the product of half the acceleration c by the square of the time t; therefore,

$$s=\frac{c}{2}t^2. \quad \ldots\ldots\ldots \quad (10)$$

Substituting this equation in equation (9), this becomes

$$pct^2=mv^2. \quad \ldots\ldots\ldots \quad (11)$$

As soon as the work has a constant value the acceleration c becomes equal to a velocity which increases from the beginning of the work during the time t from value o to value s; therefore,

$$c=\frac{s-o}{t}. \quad \ldots\ldots\ldots \quad (12)$$

Equation (11) becomes through substitution of this value,

$$pst=mv^2. \quad \ldots\ldots\ldots \quad (13)$$

Value s in this equation represents, contrary to the same denomination in equation (9), also a distance, but distance of a constant power.

Equation (13) shows the transmutation of kinetic energy into work when a moving liquid with a constant velocity v produces mechanical work.

If the work is given in kg—m—sec, then we can assume in equation (12) for $s=1$ and $t=1$, and transform it to

$$p=mv^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

If the work is given in H.P. then $pst=75N$kg.

The stated assumption represents that the pressure p is opposed to a resistance which in the time $t=1$ is capable of moving it through a distance $s=1$.

Assuming as unit of length the meter and as unit of pressure the kg., then mass m of a weight q of a cubic meter of the respective fluid divided by acceleration or

$$m=\frac{q}{g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Using these equations first for air and assuming the weight of a cubic meter of air at a temperature of 0 degree and 760 mm. barometric pressure of $q=1.29318$ kg., therefore is

$$m=\frac{1.29318}{9.81}=.13182. \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Entering this value into equation (14) we obtain as the theoretical pressure of air acting normal to a surface the relation,

$$p=.131827v^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

This theoretical formula applies only under the supposition that all molecules of air come into contact with the surface. In reality it is impossible to satisfy this assumption, since the molecules returning from the surface will collide with those arriving, and thus prevent the latter from exerting a pressure upon the surface. For this reason the numerical coefficient

in equation (17) results in practice smaller than deduced theoretically. Therefore, the latter equation is corrected by entering a constant C, which represents the value of the economical coefficient, and we have

$$p=.13182Cv^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

The value of C is always smaller than 1, and for air can be readily determined.

The pressure of air flowing perpendicularly against a plane is, as well known, $p=.123v^2$, and the relation of the two figures gives the value

$$C=\frac{.123}{.13182}=.9331. \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The diminution of pressure for air amounts to about $6\frac{3}{4}$ per cent. for the given reason.

Introducing the constant C in equation (13) and enlarging its meaning and putting it in general,

$$\frac{Co}{st}=C. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

The pressure of a fluid moving normal to a plane with a velocity v is determined from the relation

$$p=mCv^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Let us consider now inversely the case in which a plane moves with a constant volocity v against molecules at rest, so that the latter will be put in motion by the contact. The relative displacement between the molecules and the plane is in both cases alike, and therefore the resulting pressure from the motion must be the same in both cases, even if we add the assumption that the plane is normal to the direction of the motion. Therefore equation (21) can be applied generally in regard to pressure, regardless of the molecular kinetic energy or the work, which may differ in the two cases.

In the case of a moving plane against molecules at rest, v

in equation (21) represents not the velocity of the molecules after impact, but the constant velocity of the plane, or the relative displacement, and m, as before, represents the mass of the molecules, and we can therefore say $m=\frac{q}{g}$.

Applying this equation to the rotating motion of a propeller in water, the weight q of a cubic meter of water must be taken, either for fresh water or sea water, as the case may be. We will use in the equation here the latter, as this principally will have to be considered.

Assuming the mean specific weight of sea water as 1.0274, then the weight of a cubic meter

$$q=1{,}027.4 \text{ kg.}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

and the relation to the mass

$$m=\frac{1{,}027.4}{9.81}=104.7. \quad . \quad . \quad . \quad . \quad . \quad (23)$$

We substitute this value in equation (21), and it becomes then

$$p=104.7Cv^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

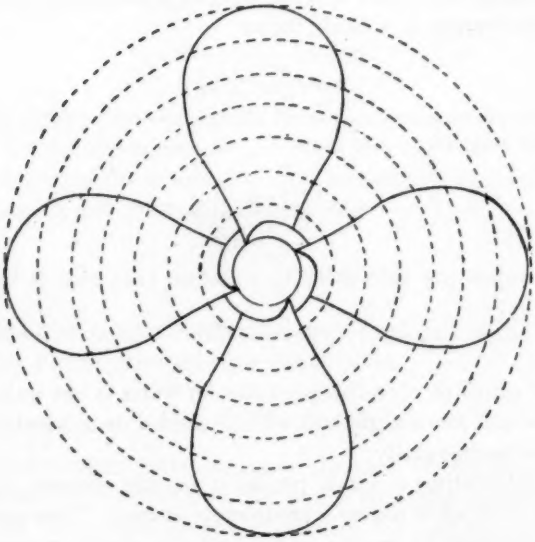
The constant C in this equation for water is yet unknown; but we will learn a method which enables us to ascertain the same experimentally.

In calculating a screw propeller we can assume that the water in which it moves is completely at rest. This condition does not exist in reality, as sea water is always more or less in motion. This motion, however, is relatively small compared with the motion which the water assumes through the rotating motion of the screw, and we shall not commit an error worth mentioning when we assume the water as at rest. The motion of the waves accelerate sometimes the speed of the vessel and sometimes they retard it, and these changeable conditions cannot be considered in the design and calculation of a propeller.

§ 3.—Axial and Tangential Pressure of a Water Wheel with Constant Pitch.

We can imagine every screw as composed of any number of concentric water wheels, with the difference between the outer and inner diameters of each wheel so small that the pitch of the same radially can be assumed as constant in cal-

Fig 3



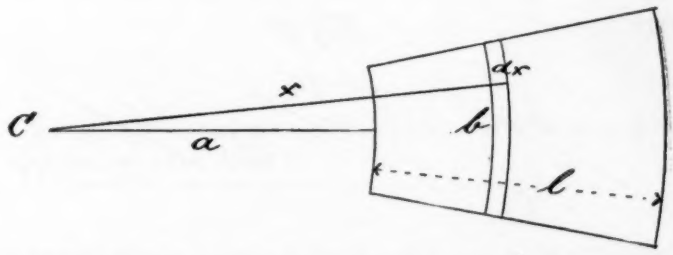
culations for practical purposes. The pitch of these single wheels may be constant or variable in the peripheral direction, but for our calculations we will assume that the pitch is constant also in the line of the periphery.

We will see directly that we can compute the work as well as the surface pressure of each water wheel with a radially-constant pitch. If we add the values found for the separate

wings we will be able to ascertain with sufficient accuracy by this the surface pressure and work of a propeller.

Let us imagine a water wheel constructed similar to the American wind wheels, with flat blades of radial and constant peripheral pitch φ, as shown in Fig. 4, which represents one

Fig 4.



blade of such a wheel. We are able to figure out the surface pressure which the wheel exerts against the water in axial direction during its revolution as follows:

At a distance x from the axis C the breadth of the blade measured on the arc is b, while the radial length is l; and let us assume also that the distance of the blade from the axis is a and that the water wheel has n blades; then the axial pressure of an infinitely small element of the blade of length dx and breadth b for all blades n by equation (8)

$$dP_4 = ndxbp \sin\varphi \cos\varphi. \quad . \quad . \quad . \quad . \quad . \quad (25)$$

For a moving plane in quiescent water we obtained (equation 21) $p = mCv^2$. The former equation by substituting this value becomes

$$dP_4 = mCnv^2 dxb \sin\varphi \cos\varphi. \quad . \quad . \quad . \quad . \quad (26)$$

The peripheral velocity v for r revolutions of the water wheel per second, at a distance x from the center, is

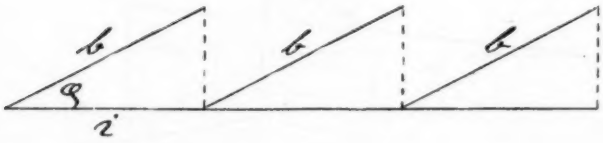
$$v = 2x\pi r, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

and our equation therefore becomes

$$dP_4 = 4mC n\pi^2 r^2 x^2 dx b \sin\varphi \cos\varphi. \quad . \quad . \quad . \quad (28)$$

The breadth b of the blade under consideration has a certain relation to the periphery of the wheel on account of the constant angle φ. Let us picture ourselves a few blades of the water wheel as in Fig. 5, seen from the periphery and

Fig 5



developed in a plane; assume, furthermore, that the wheel is so closely studded with blades that the axial projection i of each blade joins closely that of the adjoining one, then

$$b = \frac{i}{\cos\varphi}\ ; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

and as x represents any radial value,

$$i = \frac{2x\pi}{n}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

therefore,

$$b = \frac{2x\pi}{n\cos\varphi}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

Entering this value in equation (28), we obtain

$$dP_4 = 8mC\pi^3 r^2 x^3 dx \sin\varphi. \quad . \quad . \quad . \quad . \quad (32)$$

Integrating in this equation length x from the initial a to the end value $a+l$ (Fig. 4), we obtain for the total axial pressure of the water wheel the value

$$\int dPax = 8mC\pi^3 r^2 \sin\varphi \int_a^{a+l} x^3 dx$$

$$Pax = 8mC\pi^3 r^2 \sin\varphi \tfrac{1}{4}[(a+l)^4 - a^4];$$

or entering π numerically,

$$Pax = 62.01 mCr^2 \sin\varphi [(a+l)^4 - a^4]. \quad . \quad . \quad (33)$$

If in Fig. 4 we call the exterior diameter of the water wheel D_a and the interior diameter D_i, then

$$D_a = 2(a+l), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

and

$$D_i = 2a. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

Introducing the number of revolutions per minute in this equation, we have to put

$$r = \frac{T}{60}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

These values entered into equation (33) gives the total pressure in the direction of the axis of the rotating wheel of a value

$$P_{ax} = .0010766 mCT^2 \sin\varphi (D_a^4 - D_i^4). \quad . \quad . \quad (37)$$

We can deduce similarly the tangential pressure by means of equation (7).

We first obtain for the pressure the value analogous to equation (32),

$$dP_3 = 8mC\pi^3 r^2 x^3 dx \frac{\sin^2\varphi}{\cos\varphi}, \quad . \quad . \quad . \quad . \quad (38)$$

and by integration between the same limits, as before,

$$P_{tg} = 62.01 mCr^2 \frac{\sin^2\varphi}{\cos\varphi}\left[(a+l)^4 - a^4\right]. \quad . \quad . \quad . \quad (39)$$

Introducing the values of equations (34) to (36) we obtain for the total pressure of the rotating wheel in a tangential direction the equation

$$P_{tg} = 0.0010766 mCT^2 \frac{\sin^2\varphi}{\cos\varphi}\left[D_a^4 - D_i^4\right]. \quad . \quad . \quad (40)$$

Equation (37) indicates by what means we can ascertain experimentally the economical coefficient C.

A water wheel as described is made to rotate vertically in ball bearings, and movable in the same direction. Fastened to the lower end of the shaft is the water wheel, and a thread is cut on the upper end of the shaft which serves to receive weights in the shape of threaded disks. If a certain weight is lifted at the number revolutions T, then the vertical pressure P_{ax} equals the total weight G of the water wheel, inclusive of shaft and its load of disks. All values in equation (37) are therefore known excepting the coefficient C, which can be found by transposing the members

$$C=\frac{928.8G}{mT^2\sin\varphi\,(D_a^4-D_i^4)}. \quad . \quad . \quad . \quad . \quad (41)$$

The mass m will have the value $\frac{1000}{g}$ for sweet water; but for salt water the value given in equation (23) must be used.

Referring back to equation (20), $\frac{Co}{st}=C$, we see that it is immaterial in what path (s) the water is forced back by pressure p. If in reality $st \gtrless 1$, then equation (41) represents always the experimentally-obtained value of C.

§ 4.—Work of the Axial and Tangential Pressure with a Constant Pitch.

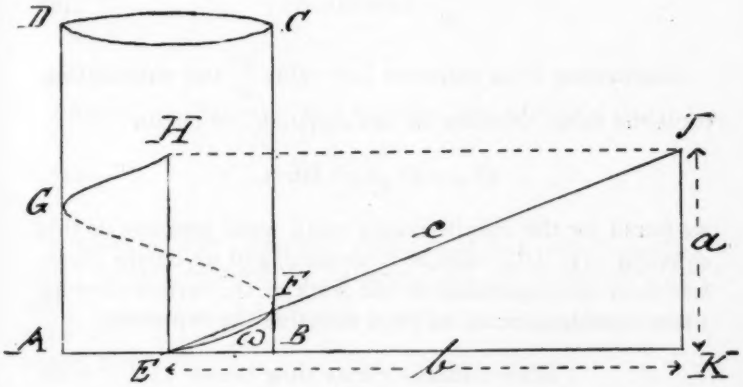
The product of force and distance gives the work.

If we imagine in a ship a water wheel of the kind described, with plane blades and constant pitch φ fitted as a propeller, and if we call the distance traveled by the wheel (neglecting slip) in an axial direction ds, in the infinitesimally small time dt, then the work per second of the infinitely small element of a blade as referred to in the previous paragraph, which is also contained in the water wheel of the vessel, is

$$dLax=dPax\frac{ds}{dt}. \quad . \quad . \quad . \quad . \quad . \quad (42)$$

Every point of the water wheel in rotating describes during the travel of the ship a cylindrical spiral, whose pitch forms an angle with the tangent of the cylinder, which we will call ω.

Fig 6



In Fig. 6 the visible surface of a cylinder is represnted by the rectangle $ABCD$, and line $EFGH$ represents the path of a spiral on the surface of this cylinder. If we imagine the path of this spiral unrolled in a straight line, then its length is given by line $EJ=c$. In the same manner we unroll the periphery of the cylinder of diameter AB in a straight line, which is represented by line $EK=b$.

The pitch of the spiral during one revolution around the cylinder is given by line $EH=KJ=a$. For every revolution of the water wheel per second the pitch is for every point of the wheel $\frac{ds}{dt}=ar$. But since we have called the angle KEJ which the spiral forms with the tangent of the cylinder ω,

$$a=b\tan\omega; \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

therefore

$$\frac{ds}{dt}=rb\tan\omega. \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

If we call the distance of the infinitesimally small surface element under consideration from the axis of rotation of the water wheel x (Fig. 4), then

$$b=2x\pi; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

therefore

$$\frac{ds}{dt}=2x\pi r \tan \omega. \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

Eliminating from equation (42) value $\frac{ds}{dt}$ and substituting for it the value found in the last equation, we obtain

$$dL_{ax}=dP_{ax}2x\pi r \tan \omega, \quad . \quad . \quad . \quad . \quad . \quad (47)$$

we found for the infinitesimally small axial pressure in this equation (32) $dP_{ax}=8mC\pi^3r^2x^3dx \sin \varphi$, and we obtain therefore from both equations for the work of the surface element under consideration in an axial direction the expression

$$dLax=16mC\pi^4r^3x^4dx \sin \varphi \tan \omega. \quad . \quad . \quad . \quad (48)$$

This equation cannot be integrated, as the angle ω has no constant value, but is a function of b and a (Fig. 6), and can not be considered constant even for a small deviation. With increasing value of x in equation (48) the angle ω grows smaller.

The work of the tangential pressure can be ascertained in a similar way to the work of the axial pressure. The tangential pressure in doing its work passes over a longer path, the further we move on a blade from the axis of rotation. The theoretical length of the path of any point of the blade for one revolution (neglecting slip) is also equal to the length of the cylindrical spiral (Fig. 6), which, unrolled, equals line c. The length of this path equals cr for r revolutions per second. Therefore the work performed by the infinitesimally small surface element per second is

$$dL_{tg}=dP_{tg}cr. \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

But as (Fig. 6)

$$c=\frac{b}{\cos\omega}, \quad \ldots\ldots\ldots \quad (50)$$

and as the considered surface element is equal to a distance x from the axis of rotation, it also is (equation 45) $b=2x\pi$, and therefore

$$c=\frac{2x\pi}{\cos\omega}. \quad \ldots\ldots\ldots \quad (51)$$

Introducing this value into equation (49) we obtain thereby

$$dL_{tg}=dP_{tg}\frac{2x\pi r}{\cos\omega}. \quad \ldots\ldots\ldots \quad (52)$$

We found for the tangential pressure of any infinitesimally small surface element by equation (38) $dP_{tg}=8mC\pi^3r^2x^3dx\frac{\sin^2\varphi}{\cos\varphi}$. Introducing this value into the former equation and noting that

$$\frac{\sin\varphi}{\cos\varphi}=\tan\varphi, \quad \ldots\ldots \quad (53)$$

we obtain thus

$$dL_{tg}=16mC\pi^4r^3x^4dx\frac{\sin\varphi\tan\varphi}{\cos\omega}. \quad \ldots \quad (54)$$

For the same reason as already mentioned for equation (48) this equation cannot be integrated.

As line a of Fig. 6 is identical with the axial advance of the propeller (neglecting slip), it is usually denominated by the letter H. We put, therefore, $H=a$.

We found (equation 43) $a=b\tan\omega$; therefore, $H=b\tan\omega$.

Furthermore (equation 45), $b=2x\pi$; and, therefore,

$$H=2x\pi\tan\omega.$$

Substituting for twice the radius x the diameter D, or $2x=D$, therefore,

$$\frac{H}{D}=\pi \tan \omega, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

which equation is well known.

§ 5.—Tangential Losses of the Water Wheel.

If we divide equation (54) by equation (48) we obtain the relation

$$\frac{dL_{tg}}{dL_{ax}}=\frac{\tan \varphi}{\cos \omega \tan \omega}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

The axial and tangential work bear to each other a certain relation which is only determined by the pitch angle φ and the spiral angle ω. But as in the propulsion of a vessel only the work L_{ax} is useful, while the work L_{tg} is a loss, it will be seen that it is important in designing a screw to make the proportion between the useless work and the total work as small as possible, or that the proportion

$$\varepsilon=\frac{dL_{tg}}{dL_{a}^{x}+dL_{tg}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

becomes a minimum.

Substituting in this equation again the values found in equations (48) and (54) for the work and expressing this in per cent.,

$$\varepsilon=\frac{100}{\frac{\cos \omega \tan \omega}{\tan \varphi}+1} \text{ per cent.};$$

or, since $\tan \omega=\frac{\sin \omega}{\cos \omega}$,

$$\varepsilon=\frac{100}{1+\frac{\sin \omega}{\tan \omega}} \text{ per cent.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

The tangential loss in a screw propeller is, as seen from equations (56) and (58), independent of the revolutions and only determined by the angles φ and ω. This loss given in per cent. of the total work by equation (58) applies not only to a single surface element, but to the whole wheel, but for

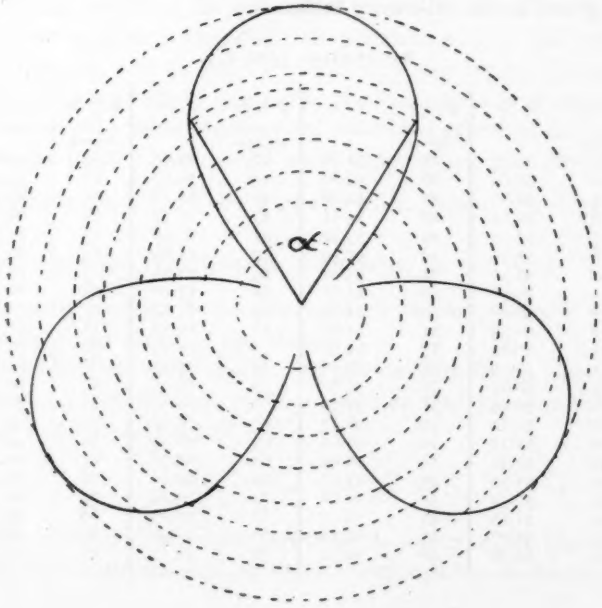
any propeller with a peripheral constant pitch angle φ and a peripheral constant spiral angle ω. To form an idea of the magnitude of the tangential losses the following table gives the values ε calculated for a few examples of the respective angles of φ and ω.

TANGENTIAL LOSS ε PER CENT.

φ	5°	10°	15°	20°	25°	30°	35°
ω= 5°	50.10	66.92	75.45	80.67	84.25	86.88	88.93
10°	33.5	50.38	60.68	67.7	72.86	76.88	80.13
15°	25.26	40.52	50.87	58.44	64.31	69.05	73.02
20°	20.37	34.01	43.93	51.55	57.69	62.80	67.18
25°	17.15	29.44	38.81	46.28	52.46	57.74	62.36
30°	14.89	26.07	34.89	42.12	48.26	53.59	58.34
35°	13.24	23.51	31.84	38.82	44.84	50.16	54.97

The table shows that the tangential loss amounts always to more than 50 per cent. when the pitch angle φ is greater than

Fig 7



the spiral angle ω. Inversely, if φ is smaller than ω the tangential loss becomes smaller than 50 per cent.; but in this case the ship cannot make the prescribed speed. Unfavorable conditions arise in both cases, either when φ is greater than ω or when φ is smaller than ω. The most advantageous degree of efficiency is obtained when φ equals ω, in which case between angle 5 degrees and angle 35 degrees a tangential loss of from 50.1 per cent. to 54.97 per cent. is unavoidable. Therefore in all our further calculations we assume $\varphi=\omega$, which transforms equation (58) into

$$\varepsilon=\frac{100}{1+\frac{\sin\varphi}{\tan\varphi}} \text{ per cent.,}$$

or

$$\varepsilon=\frac{100}{1+\cos\varphi} \text{ per cent.} \quad . \quad . \quad . \quad . \quad . \quad (59)$$

The proportionate values of ε of angle φ as per equation (59) are given in the following table:

TANGENTIAL LOSS ε %.

$\varphi=\omega$	ε %	$\varphi=\omega$	ε %	$\varphi=\omega$	ε %	$\varphi=\omega$	ε %
Degrees.		*Degrees.*		*Degrees.*		*Degrees.*	
3	50.03	25	52.46	47	59.45	69	73.61
4	50.06	26	55.66	48	59.91	70	74.52
5	50.10	27	52.88	49	60.38	71	75.44
6	50.14	28	53.11	50	60.87	72	76.39
7	50.19	29	53.34	51	61.38	73	77.37
8	50.24	30	53.59	52	61.89	74	78.39
9	50.31	31	53.84	53	62.43	75	79.44
10	50.38	32	54.11	54	62.98	76	80.52
11	50.46	33	54.39	55	63.55	77	81.63
12	50.55	34	54.68	56	64.14	78	82.79
13	50.65	35	54.97	57	64.74	79	83.97
14	50.75	36	55.28	58	65.36	80	85.20
15	50.87	37	55.60	59	66.01	81	86.47
16	50.99	38	55.93	60	66.67	82	87.78
17	51.12	39	56.27	61	67.35	83	89.14
18	51.25	40	56.62	62	68.05	84	90.54
19	51.40	41	56.99	63	68.78	85	91.98
20	51.55	42	57.37	64	69.52	86	93.48
21	51.72	43	57.76	65	70.29	87	95.03
22	51.89	44	58.16	66	71.09	88	96.63
23	52.07	45	58.58	67	71.91	89	98.28
24	52.26	46	59.01	68	72.75	90	100.00

Screws with peripherally varying pitch are like those in which φ is either greater or smaller than ω; they have also a lower efficiency than screws in which φ is equal ω.

We can understand now the cause of cavitations observed in screw propellers of fast-running turbines, for if angle φ is larger than ω the vessel cannot follow as fast as the large propeller pitch travels, which latter was based upon a too large angle φ, and the available axial pressure of the screw, as well as the power of the engines, are not sufficient for such a large pitch. Furthermore, it requires time for the water to fill the void produced by the rotating screw; the rotation of the screw is too rapid for the water to follow.

§ 6.—Power Consumption of the Water Wheel.

By equation (48), which cannot be integrated, we determined the work produced by the water wheel in an axial direction.

We can ascertain the amount of this work in still another way, not only for a single surface element, but for the whole water wheel. As we have seen just now, the tangential work always bears a certain relation to the axial work, and thus we have a way to calculate the total work of a screw propeller.

Every point of the water wheel travels in axial direction over the path $\frac{ds}{dt}$; the work of the axial thrust, if a vacuum existed behind the back of the blades, is, therefore, $P_{ax}\frac{ds}{dt}$. But there is no vacuum, as water follows and exerts a pressure upon the backs of the blades.

If we call the axial component of this pressure p_{ax} then the pressure P_{ax} is reduced by that amount, and the remaining work for the axial thrust is

$$L_{ax} = (P_{ax} - p_{ax})\frac{ds}{dt} \quad . \quad . \quad . \quad . \quad . \quad (60)$$

The motion of the water towards the backs of the blades is to fill the void produced by the rotating motion of the screw.

The direction of motion of the following water is so manifold that probably all directions in space are represented. Therefore we will assume that the water follows in three directions perpendicular to each other. We will imagine, therefore, through a point in space a right-angled tri-axial system of coördinates whose z axis runs parallel with the axis of motion of the screw; then, according to this simplified idea, one-third of all water molecules flow in the direction of the x axis, one-third in the direction of the y axis and one-third in the direction of the z axis into any void formed by the rotating water wheel. The molecules moving parallel with the x and y axes cause no change in the specific surface pressure, while the molecules moving in the direction of the z axis remain behind, and never reach the screw propeller. Therefore the pressure of the water flowing into the void behind the propeller is composed only of molecules parallel with the x and y axes, or only two-thirds of all molecules flowing after. Therefore in the last equation we have to make $p_{a_x}=\frac{2}{3}P_{ax}$, which transforms it into

$$L_{ax}=\tfrac{1}{3}P_{ax}\frac{ds}{dt}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

For the tangential work we found equation (56), in which we make $\omega=\varphi$, $L_{tg}=\dfrac{L_{ax}}{\cos\varphi}$; and, therefore,

$$L_{tg}=\frac{P_{ax}}{3\cos\varphi}\frac{ds}{dt}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (62)$$

If we name the total value of both with λ or the sum

$$\lambda=L_{ax}+L_{tg} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (63)$$

the addition of both (61) and (62) gives

$$\lambda=\frac{P_{ax}}{3}\left[1+\frac{1}{\cos\varphi}\right]\frac{ds}{dt}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

By equation (59) $\cos\varphi=\dfrac{100}{\varepsilon}-1$, and our equation becomes after entering this value

$$\lambda=\frac{P_{ax}}{3\,(1-.01\,\varepsilon)}\frac{ds}{dt}. \quad . \quad . \quad . \quad . \quad . \quad (65)$$

The value of the total pressure in the direction of the axis is determined exactly by equation (37). If we enter this value in the last equation we obtain for the total work of a water wheel with constant pitch the relation

$$\lambda=.0003589\ mCT^2\frac{\sin\varphi}{1-.01\,\varepsilon}(D_a^4-D_i^4)\frac{ds}{dt}. \quad . \quad . \quad (66)$$

From this equation we can ascertain at once the horse-power N necessary to drive the wheel.

As $N=\frac{\lambda}{75}$, therefore

$$N=4.785\times 10^{-6}mCT^2\frac{\sin\varphi}{1-.01\,\varepsilon}(D_a^4-D_i^4)\frac{ds}{dt}. \quad . \quad (67)$$

The two equations (66) and (67) apply only within those limits in which the value ε is nearly constant, and represent the mean value of the values belonging to the diameters D_a and D_i.

§7. Determination of Thrust and Power of the Screw Propeller.

The equations given in the last paragraph were deduced under the supposition that the single blades of the respective wheel were placed close to each other (Fig. 5) and that the water could only pass through in a tangential direction but not in the direction of the axis.

If we use instead of the complete wheel only a half, quarter or n times segment of the same, then the thrust which is exerted in the axial and tangential direction will be only one-half, one-quarter or n times that of the whole wheel. The axial thrust of the same is therefore

$$P_a=nP_{ax}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

and the tangential thrust

$$P_t = nP_{tp}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

In like manner we find the power necessary in horsepower from equation (67)

$$N = 4.785 \times 10^{-6}\, mCnT^2 \frac{\sin\varphi}{1-.01\,\varepsilon}(D_a^4 - D_i^4)\frac{ds}{dt}. \quad . \quad (70)$$

If in the two previous equations we enter the values for the thrust from equations (37) and (40), and also observe that the water flowing in behind will reduce the active thrust to one-third that of the theoretical (see equations (60) and (61)), then we obtain

$$P_a = .0003589\, mCnT^2 \sin\varphi\,(D_a^4 - D_i^4); \quad . \quad . \quad (71)$$

$$P_t = .0003589\, mCnT^2 \frac{\sin^2\varphi}{\cos\varphi}(D_a^4 - D_i^4). \quad . \quad . \quad (72)$$

We can, as already stated in § 3, imagine each screw propeller with a constant peripheral and radially-variable pitch, divided into any number concentric water wheels with a constant peripheral and radial pitch. The blades of the propeller are thus dissected into annular segments, and each will be n times that of the whole ring or annulus.

If the axial projection of an annular segment includes the angle a from the center, and if the number of blades is Z, then

$$n = \frac{Za}{360}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (73)$$

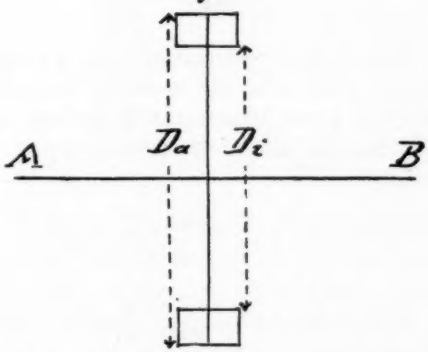
The calculation for thrust and power required by a given screw is therefore not difficult. By means of the above equations we can design a new propeller, as will be shown further on.

In deducing all the former equations it was assumed that each cross section of a blade was perpendicular to the axis of

revolution of the screw. Such a cross section is represented by Fig. 8.

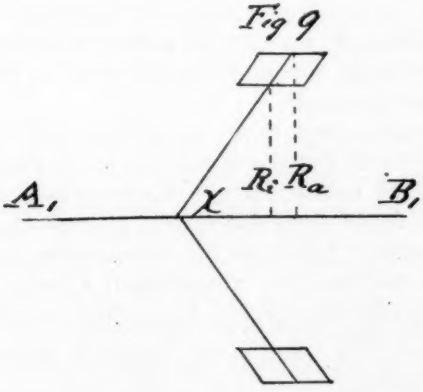
By line AB we show the axis of revolution of a screw propeller, and the small rectangles give the side view of one annular segment, as mentioned before. Line D_i stands for the inner and line D_a for the outer diameter of the respective ring.

Fig 8



In Fig. 8 all annular segments were arranged so that their connecting lines were perpendicular to axis AB, while in Fig. 9 we show an arrangement in which the connecting

Fig 9



lines of the segments form an angle with axis A_1B_1 which we will name χ. The Niki and Zeise propellers are formed in the latter manner.

If we name the vertical radii (Fig. 9) R_i and R_a, then

$$D_i=\frac{2\,R_i}{\sin\chi}; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

$$D_a=\frac{2\,R_a}{\sin\chi}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (75)$$

Entering these values in equation (70), (71) and (72), and substituting also mass m for salt water by the value of equation (23) $m=104.7$, we obtain the following relations, in which to eradicate the sum of the separate rings we put $P=\Sigma dP$; therefore

$$\Sigma dP_a=.6012\ CnT^2\frac{\sin\varphi}{\sin^4\chi}(R_a^4-R_i^4); \quad . \quad . \quad . \quad (76)$$

$$\Sigma dP_i=.6012\ CnT^2\frac{\sin^2\varphi}{\cos\varphi\sin^4\chi}(R_a^4-R_i^4); \quad . \quad . \quad (77)$$

$$\Sigma dN=.0127\frac{CnT^2\sin\varphi}{(1-.01\,\varepsilon)\sin^4\chi}(R_a^4-R_i^4)\frac{ds}{dt}. \quad . \quad (78)$$

The computations for a propeller of this kind also offer no difficulties.

As the differences between the powers of the radii occur in every computation of a propeller, the table on the following page has been arranged.

It is necessary here to say something about the popular fallacy of the centrifugal action of the screw propeller. The water is forced outward partly in a tangential direction by the rotation of the screw and thereby gets out of active range of the propeller. Therefore the water cannot be set in centrifugal motion and also cannot exert a centrifugal force. There exists only a tangential force which presses the water outward.

TABLE FOR $R_a^4 - R_i^4$.

R_a	R_i	$R_a^4 - R_i^4$	R_a	R_i	$R_a^4 - R_i^4$
0.2	0.1	0.0006	1.9	1.8	2.535
0.3	0.2	0.0065	2.0	1.9	2.968
0.4	0.3	0.0175	2.1	2.0	3.448
0.5	0.4	0.0369	2.2	2.1	3.977
0.6	0.5	0.0671	2.3	2.2	4.559
0.7	0.6	0.1105	2.4	2.3	5.193
0.8	0.7	0.1695	2.5	2.4	5.886
0.9	0.8	0.2465	2.6	2.5	6.633
1.0	0.9	0.3439	2.7	2.6	7.446
1.1	1.0	0.4640	2.8	2.7	8.325
1.2	1.1	0.6096	2.9	2.8	9.263
1.3	1.2	0.7824	3.0	2.9	10.270
1.4	1.3	0 9857	3.1	3.0	11.350
1.5	1.4	1.2208	3.2	3.1	12.507
1.6	1.5	1.4912	3.3	3.2	13.733
1.7	1.6	1.7985	3.4	3.3	15.044
1.8	1.7	2.1448	3.5	3.4	16.436

§ 8.—Power Consumption of the Vessel.

Of the different resistances opposed to the motion of a vessel that of the water is the most formidable. In the second line stands the frictional resistance of the vessel in the water, while in the third line we must consider the air resistance of that part of the vessel protruding from the water.

The latter two resistances differ according to the kind of vessel; we express them, therefore, in fractions of the water resistance, or rather multiply the water resistance by the coefficient K, which shall include the addition of all resistances. We put, therefore,

$$K = \frac{W_e}{W}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (79)$$

and express thus the relation between the theoretical water resistance W and the sum of all effective resistances W_e.

The resistance of the water is equal to the area of midship section measured at the maximum cross section (Q) of the vessel multiplied by the specific pressure (p) of the water against this area. Near the surface this pressure is small, but increases rapidly with the depth of the midship section. We give this resistance, therefore, by the sum

$$W = \Sigma dQdp. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (80)$$

In connection with the former equation we obtain the value of the effective water pressure

$$W_e = K\Sigma dQdp. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

The work required in moving a vessel when towing it is composed of the resistance W_e moved in each second through the space u, which corresponds with the effective speed of the vessel. The necessary work required for this per second, equal to the axial thrust of the screw, is expressed by the equation

$$L_a = W_e u$$

or

$$L_a = Ku\Sigma dQdp. \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

The pressure of the water is the same on both sides of the immersed midship section as long as the vessel is not moving, but after beginning to move this pressure is different, and the question is how great this pressure is either in front of the vessel or behind it. If we designate the former by dp_1 and the latter by dp_2, then the actual pressure of the water during the motion of the vessel is

$$dp = dp_1 - dp_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (83)$$

The motion of the water behind the vessel is the same as ascertained in § 6 for the water screw.

Here again let us imagine that the molecules of water flow in three directions perpendicular to each other into the void produced by the moving vessel, and that the z axis coincides with the direction of the ship, then only the molecules from the x and y axes can reach the vessel, while the molecules from the z axis are unable to reach the ship, and therefore cannot exert a pressure either.

Our simplified idea of the suction is easier understood when considered with a vessel than with rotating propeller; both

actions result from the same cause, and therefore the dynamic effect must be the same.

Similarly to the rotating propeller we must in case of a moving vessel make $dp_2=\frac{2}{3}dp_1$, and thereby the last equation becomes

$$dp=\tfrac{1}{3}dp_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (84)$$

Entering this value into equation (82) we thereby obtain

$$L_a=\tfrac{1}{3}Ku\Sigma dQdp_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

For the pressure dp_1 we substitute the height of the water column h in m. For salt water we found (equation 22) the weight of a cubic meter to be 1,027.4 kg., and, therefore, $dp_1=1{,}027.4\,dh$. This value entered in equation (85) makes

$$L_a = 342.45\,Ku\Sigma dQdh. \quad . \quad . \quad . \quad . \quad . \quad (86)$$

The velocity u in this equation is in meters per second, and can be ascertained in sea miles or knots from their relation. A knot (V) means the distance of 1,852 m. per hour, therefore

$$W=\frac{1{,}852V}{3{,}600}=.5144V. \quad . \quad . \quad . \quad . \quad . \quad (87)$$

Entering this value in the former equation, we get for the work of towing the ship the value

$$L_a=176.17\,KV\Sigma dQdh. \quad . \quad . \quad . \quad . \quad . \quad (88)$$

The effective speed u in equation (87) is identical with the theoretical speed as defined in § 4, $\frac{ds}{dt}$ less the retardation produced by the resistance represented by the coefficient K. The relation of this retardation to the theoretical speed is the slip of the propeller. The slip percentage is, therefore,

$$S=\frac{\left(\frac{ds}{dt}-u\right)}{\frac{ds}{dt}}\text{per cent.,} \quad . \quad . \quad . \quad . \quad . \quad (89)$$

and therefore,

$$u=(1-0.01\,S)\frac{ds}{dt}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (90)$$

The relation between the two velocities is nothing else than the coefficient K; for

$$W_e\,u=W\frac{ds}{dt}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (91)$$

and it follows from equation (79) that

$$K=\frac{\frac{ds}{dt}}{u}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

This equation gives with equation (90) the value

$$K=\frac{1}{1-0.01\,S}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (93)$$

Eliminating from equation (88) quantity K, by this value we obtain

$$L_a=\frac{176.17\,V}{1-0.001\,S}\Sigma dQdh. \quad . \quad . \quad . \quad . \quad (94)$$

Referring back to equation (56), in which we assumed $\varphi=\omega$, therefore $L_t=\frac{L_a}{\cos\varphi}$, or with value of equation (94)

$$L_t=\frac{176.17\,V}{1-.01S}\times\frac{\Sigma dQdh}{\cos\varphi}. \quad . \quad . \quad . \quad . \quad . \quad (95)$$

The total work λ is again (equation 63) $\lambda=L_a+L_t$, and after entering above values

$$\lambda=\left(1+\frac{1}{\cos\varphi}\right)\frac{176.17V}{1-.01S}\Sigma dQdh. \quad . \quad . \quad . \quad (96)$$

From equation (59) we obtain $\cos\varphi=\frac{100}{\varepsilon}-1$, and this value in connection with the former equation gives

$$\lambda=\frac{176.17V}{1-.01S}\times\frac{\Sigma dQdh}{1-.01\varepsilon}. \quad . \quad . \quad . \quad . \quad (97)$$

The work in effective horsepower is $N_e=\frac{\lambda}{75}$; therefore

$$N_e=\frac{2.349V}{1-.01S}\times\frac{\Sigma dQdh}{1-.01\varepsilon}. \quad . \quad . \quad . \quad . \quad (98)$$

This equation permits the immediate calculation of the required horsepower from the immersed midship section and the speed of the ship, by assuming the most probable values for the slip S and the mean tangential loss ε.

This equation (98) has no resemblance to the so-called Admiralty formula. Nothing is found in it that could justify the cube of speed in the latter.

§ 9.—Determination of Pitch Angle.

Transforming the members of equation (90), we obtain $\frac{ds}{dt}=\frac{u}{1-0.01S}$. Eliminating quantity u by means of equation (87), we find

$$\frac{ds}{dt}=\frac{0.5144V}{1-0.01S}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (99)$$

We found before the theoretical speed (equation 46) $\frac{ds}{dt}=2x\pi r\tan\omega$. From the two equations we get, therefore,

$$\tan\omega=\frac{0.2572V}{x\pi r(1-0.01S)}. \quad . \quad . \quad . \quad . \quad (100)$$

But (equation 36) $r=\frac{T}{60}$, and from Fig. 9, $x=\frac{R}{\sin\chi}$, if R represents the mean value between the inner and outer radius of a ring, as in Fig. 9. These two values entered into equation (100), and π given its numerical value, gives

$$\tan\omega=\frac{4.912V\sin\chi}{RT(1-.01S)}. \quad . \quad . \quad . \quad . \quad (101)$$

As explained in § 5, the pitch angle φ must be made equal to the spiral angle ω; for this reason, the same equation applies to the pitch angle

$$\tan\varphi=\frac{4.912V\sin\chi}{RT(1-.01S)}. \quad . \quad . \quad . \quad . \quad (102)$$

For each single ring we will have to determine the mean value of φ as per Fig. 7.

Example I.

To understand the deduced equations better, the same will be used for the calculations of a propeller. A fast steamer of 2,200 tons displacement is to have a speed V of 22 knots. The slip may be assumed, according to experimental formula, to 17 per cent.

By comparison with known steamers we will probably require 7,000—8,000 effective horsepower. We consider, therefore, that we need two turbines with $T=230$ revolutions per minute.

We determine first the pitch angle by equation (102).

$$\tan\varphi=\frac{4.912V\sin\chi}{RT(1-.01S)}.$$

For the intended screw we assume angle $\chi=90$ degrees (Fig. 9), and obtain $\tan\varphi=\frac{4.912\times22}{R\,230\times.83}=\frac{.5661}{R}$.

By means of this equation we determine the pitch angles

of the single rings of Fig. 7. Increasing the diameter of the rings by .2 m. we obtain the values in the following table. The values of ε are taken from table §5.

D_a	D_i	R (mean)	φ	$\varepsilon\,\%$
0.4	0.2	0.15	75° 9′	79.5
0.6	0.4	0.25	66 10	71.2
0.8	0.6	0.35	58 17	65.6
1.0	0.8	0.45	51 31	61.6
1.2	1.0	0.55	45 50	58.9
1.4	1.2	0.65	41 3	57.0
1.6	1.4	0.75	37 3	55.6
1.8	1.6	0.85	33 40	54.6
2.0	1.8	0.95	30 47	53.8
2.2	2.0	1.05	28 20	53.2
2.4	2.2	1.15	26 12	52.7
2.6	2.4	1.25	24 22	52.3
2.8	2.6	1.35	22 45	52.0

For the calculation of the horsepower for the engine the mean value of ε can be taken at 53 per cent., as will appear later from the table for $\frac{\Sigma dN}{n}$.

The horsepower needed can be ascertained from equation (98)

$$N_e = \frac{2.349 V}{1 - .01 S} \times \frac{\Sigma dQdh}{1 - .01 \varepsilon}.$$

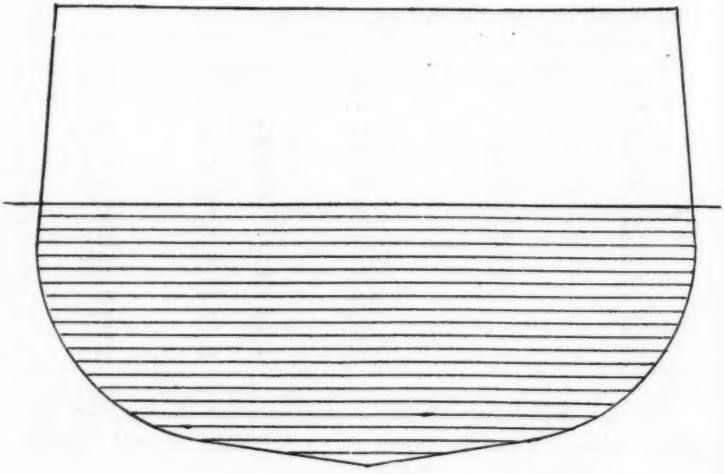
By using the three known values we have

$$N_e = \frac{2.349 \times 22\ \Sigma dQdh}{.83 \times .47}$$

$$N_e = 132.47 \Sigma dQdh.$$

The summation $\Sigma dQdh$ represents the sum of all infinitesimally small elements of the cross section, each multiplied by the height of the column of water above it in feet. Fig. 10 represents the midship-section area to a scale of 1 to 100, and if the submerged part of the cross section is divided into strips .2 m. wide, which is sufficiently accurate, then the mean height dh of each strip is as .1, .3, .5 . . . m.

Fig 10



From the drawing we obtain at once by measurement:

.....	dQ	dh	$dQdh$
1st strip	$10.00 \times .2 = 2.00$ m²	0.1	0 200
2d	$10.05 \times .2 = 2.01$	0.3	0.603
3d	$10.05 \times .2 = 2.01$	0.5	1.005
4th	$10.00 \times .2 = 2.00$	0.7	1.400
5th	$10.00 \times .2 = 2.00$	0.9	1.800
6th	$9.95 \times .2 = 1.99$	1.1	2.189
7th	$9.90 \times .2 = 1.98$	1.3	2.574
8th	$9.80 \times .2 = 1.96$	1.5	2.940
9th	$9.70 \times .2 = 1.94$	1.7	3.298
10th	$9.50 \times .2 = 1.90$	1.9	3.610
11th	$9.30 \times .2 = 1 86$	2.1	3.906
12th	$9.00 \times .2 = 1.80$	2.3	4.140
13th	$8.70 \times .2 = 1.74$	2.5	4.350
14th	$8.35 \times .2 = 1.67$	2.7	4.509
15th	$7.90 \times .2 = 1.58$	2.9	4.582
16th	$7.40 \times .2 = 1.48$	3.1	4.588
17th	$6.40 \times .2 = 1.28$	3.3	4.224
18th	$5.60 \times .2 = 1.12$	3.5	3.920
19th	$3.60 \times .2 = 0.72$	3.7	2.664
20th	$1.00 \times .2 = 0.20$	3.9	0.780
	33.24 m²		57.282

Introducing this value into the last equation we obtain for the required indicated horsepower the number

$$N_e = 132.47 \times 57.282 = 7{,}588 \text{ H.P.}$$

We require, therefore, two turbines each of 3,800 H.P.

The theoretical speed can be ascertained from equation (99)

$$\frac{ds}{dt} = \frac{.5144\,V}{1 - .01\,S}$$

$$\frac{ds}{dt} = \frac{.5144 \times 22}{.83} = 13.635 \text{ m. sec.}$$

The obtained values are used to build up a propeller from equation (78)

$$\Sigma dN = .0127 \frac{C n T^2 \sin\varphi}{(1 - .01\,\varepsilon)\sin^4\chi}(R_a^4 - R_i^4)\frac{ds}{dt}$$

of as many rings until we have the required engine power of 3,800 H.P. for each of the two propellers. We have assumed $\chi = 90$ degrees and $T = 230$; the economic coefficient C for water is still an unknown quantity, but is probably not far from 0.5. It was explained in § 2 that the molecules receding from the plane meet those moving towards the plane and prevent the latter from exerting a pressure upon the same. The loss from this cause was found to amount for air to $6\frac{2}{3}$ per cent. Water, on account of its greater density, would probably only allow one-half of the molecules moving towards the plane to reach the same, the other half is prevented from reaching the plane by the receding molecules.

To illustrate the process of calculation we assume the value of C as .60.

The above equation can be simplified by substituting the known values

$$\Sigma dN = .0127 \times .60\; n\; 230^2 \frac{13.635 \sin\varphi}{1 - .01\,\varepsilon}(R_a^4 - R_i^4),$$

$$\Sigma dN = \frac{5{,}496\, n \sin\varphi}{1 - .01\,\varepsilon}(R_a^4 - R_i^4),$$

and, as the value for n is uncertain at present,

$$\frac{\Sigma dN}{n}=\frac{5.496 \sin \varphi}{1-.01\,\varepsilon}(R_a^4-R_i^4).$$

Reserving for the hub the value $D=.4$ m., by means of this equation and use of table of § 7 we obtain the following rings:

R_a	R_i	φ	ε %	$\frac{\Sigma dN}{n}$
0.3	0.2	66° 10′	71.2	113
0.4	0.3	58 17	65.6	238
0.5	0.4	51 31	61.6	414
0.6	0.5	45 50	58.9	644
0.7	0.6	41 3	57.0	928
0.8	0.7	37 3	55.6	1,265
0.9	0.8	33 40	54.6	1,655
1.0	0.9	30 47	53.8	2,094
1.1	1.0	28 20	53.2	2,586
1.2	1.1	26 12	52.7	3,127
1.3	1.2	24 22	52.3	3,719
1.4	1.3	22 45	52.0	4,365
				21,148

The total projected surface of all propeller rings gives 21,148 H.P. We need only 3,800, therefore

$$\frac{3,800}{21,148}\times 100=17.97 \text{ per cent.}$$

of the annular surface must be distributed for the values of n.

The value of n was by equation (73)

$$n=\frac{Z_a}{360}$$

determined, in which Z represents the number of blades. If we choose for our intended propeller three blades, then we must make

$$n=\frac{3a}{360}=\frac{a}{120}$$

and the mean value of a is

$$120 \times \frac{17.97}{100} \text{ or about 22 degrees.}$$

We now make a design as shown by Fig. 11 with this angle, and ascertain by trial suitable values for n. Thus we obtain the following values:

R_a	R_i	$\frac{\Sigma dN}{n}$	a	n	ΣdN
0.3	0.2	113	45°	0.3750	43
0.4	0.3	238	42	0.3500	83
0.5	0.4	414	39	0.3250	134
0.6	0.5	644	36	0.3000	193
0.7	0.6	928	34	0.2833	263
0.8	0.7	1,265	32	0.2667	337
0.9	0.8	1,655	30	0.2500	414
1.0	0.9	2,094	28	0.2333	489
1.1	1.0	2,586	26	0.2167	560
1.2	1.1	3,127	22	0.1833	573
1.3	1.2	3,719	17	0.1417	527
1.4	1.3	4,365	9	0.0750	327
					3,943 H.P.

The power needed for the propeller is somewhat larger than intended, and we therefore make each engine of about 3,950 H.P., especially since we did not consider in our first estimate the power needed for the hubs, which likewise as the immersed midship section, require power for their axial propulsion through the water.

The width of the blades on the arcs of the center of the rings can be determined from the drawing by dividing these arcs by cos φ, as given by the axial projection of the drawing.

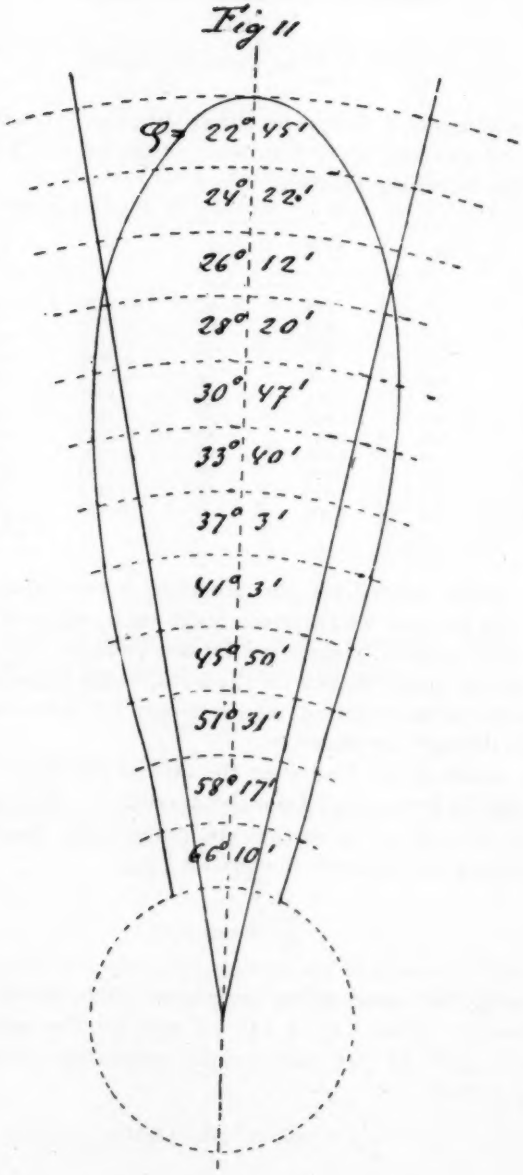
Applying our example to equation (55),

$$\frac{H}{D} = \pi \tan \varphi,$$

and using the value of the peripheral pitch, which for the middle-ring radius 1.25 is 24° 22′ and for the middle-ring radius 1.35 is 22° 45′ and for the outer-ring radius 1.4 is 21° 56′, then

$$\frac{H}{D} = \pi \tan 21° 56' = 1{,}265.$$

Fig 11



This value corresponds well with conditions of common practice. The peripheral velocity of the blades figures out to 33.72 m. sec.

Example II.

To be able to make comparisons between a turbine steamer and a piston-engine vessel we take the same steamer of 2,200 tons displacement and 22 knots speed, which we figured out before, and assume that it has two reciprocating engines making 80 revolutions. The slip must be again taken to 17 per cent. for sake of comparison.

The pitch angles by equation (102) will be

$$\tan\varphi=\frac{4.912\,V\sin\chi}{RT(1-.01\,S)}.$$

By substituting the known values we obtain

$$\tan\varphi=\frac{4.912\times 22}{R\times 80\times .83}=\frac{1.6275}{R}.$$

For the angles we get the following values:

D_a	D_i	R (mean)	φ		ε %
0.4	0.2	0.15	84°	44′	93.1
0.6	0.4	0.25	81	16	86.8
0.8	0.6	0.35	77	52	82.7
1.0	0.8	0.45	74	33	78.9
1.2	1.0	0.55	71	20	75.8
1.4	1.2	0.65	68	14	73.0
1.6	1.4	0.75	65	15	70.5
1.8	1.6	0.85	62	25	68.3
2.0	1.8	0.95	59	43	66.5
2.2	2.0	1.05	57	10	64.8
2.4	2.2	1.15	54	45	63.4
2.6	2.4	1.25	52	28	62.2
2 8	2.6	1.35	50	19	61.0
3.0	2.8	1.45	48	18	60.0
3.2	3.0	1.55	46	24	59.2
3.4	3.2	1.65	44	37	58.4
3.6	3.4	1.75	42	55	57.7
3.8	3.6	1.85	41	20	57.0
4.0	3.8	1.95	39	51	56.6
4.2	4.0	2.05	38	27	56.1
4.4	4.2	2.15	37	7	55.7

We can take the mean value of ε for the calculation of the horsepower to 58 per cent., as can be seen from the table under $\frac{\Sigma dN}{n}$.

Equation (98) gives for the necessary horsepower

$$N_e = \frac{2.349 V}{1 - .01 S} \times \frac{\Sigma dQdh}{1 - .01 \varepsilon},$$

the value

$$N_e = \frac{2.349 \times 22 \times 57.282}{.83 \times .42} = 8{,}491 \text{ H.P.}$$

The power consumption is considerably greater with the lower number revolutions than before. We need two engines of 4,250 H.P. each.

The theoretical speed is the same as before,

$$\frac{ds}{dt} = 13.635.$$

Building up a propeller by equation (78)

$$\Sigma dN = .0127 \frac{CnT^2 \sin\varphi}{(1 - .01\,\varepsilon)\sin^4\chi}(R_a^4 - R_i^4)\frac{ds}{dt}.$$

From this we get

$$\frac{\Sigma dN}{n} = .0127 \frac{.60 \times .80^2 \times 13.635}{1 - .01\,\varepsilon} \sin\varphi\,(R_a^4 - R_i^4)$$

and

$$\frac{\Sigma dN}{n} = \frac{665 \sin\varphi}{1 - .01\,\varepsilon}(R_a^4 - R_i^4).$$

By means of this equation we obtain for our propeller the following values, but we must in arranging these reserve more room for the hub, on account of the lesser number of revolutions. We therefore reserve for the hub $D = .6$ m.

R_a	R_i	φ	ε %	$\frac{\Sigma dN}{n}$
0.4	0.3	77° 52′	82.7	66
0.5	0.4	74 33	78.9	112
0.6	0.5	71 20	75.8	175
0.7	0.6	68 14	73.0	253
0.8	0.7	65 15	70.5	347
0.9	0.8	62 25	68.3	458
1.0	0.9	59 43	66.5	590
1.1	1.0	57 10	64.8	737
1.2	1.1	54 45	63.4	905
1.3	1.2	52 28	62.2	1,092
1.4	1.3	50 19	61.0	1,294
1.5	1.4	48 18	60.0	1,516
1.6	1.5	46 24	59.2	1,760
1.7	1.6	44 37	58.4	2,019
1.8	1.7	42 55	57.7	2,296
1.9	1.8	41 20	57.0	2,589
2.0	1.9	39 51	56.6	2,914
2.1	2.0	38 27	56.1	3,248
				22,371

Only 4,250 of these 22,371 horsepower will be used; we must distribute, therefore,

$$\frac{4{,}250}{22{,}371}\times 100 = 19 \text{ per cent.}$$

of the annular surfaces upon the value n.

It will be unnecessary to make a drawing in this case, as this would offer nothing new, and would be similar to drawing 11, only somewhat larger.

It will be of interest to ascertain the relation of equation (55)

$$\frac{H}{D} = \pi \tan \varphi.$$

Angle φ of the extreme periphery is found to be 39° 9′; therefore

$$\frac{H}{D} = \pi \tan 39° 9' = 2.558,$$

a value much greater than the one obtained for the first example.

CONCLUSION.

The two examples selected can be compared only with each other, not with results from praxis, as we assumed for the constant C the haphazard value of 0.6, and this in reality is hardly sufficiently exact. As already mentioned, the value C has to be ascertained experimentally as per equation (41) before the given equations are applicable for praxis.

The value $\frac{H}{D}$ results in our second example larger than is usual in praxis, and this is a defect of the former method of calculation, that one of the most important values is fixed at the beginning, without knowing if the assumed proportion corresponds approximately to reality. If this relation is fixed at the start, then we are compelled to make the diameter unnecessarily large, to suit the accepted relation. The work of the propeller increases with its diameter. The power needed for the propulsion of the vessel is fixed. Any unnecessary increase in the diameter of the screws subtracts from the neighborhood of the hub work, which is transferred to the increase of diameter. The pitch angles near the hub must be kept so small that the water can exert no pressure in the direction of the axis.

A second very important fact is proven by theory, that the power for the vessel (equation 98) differs as the revolutions of the engine. The slower running the engine the greater must be the pitch angle φ to attain the required speed; but with an increase in the angle φ the tangential losses increase without any gain to the axial thrust, and thus is seen clearly the peculiarity of the theory. The fast-running turbines for this reason save considerable power when compared with slow-rotating reciprocating engines, as seen from our examples, where the power needed drops from 4,240 H.P. to 3,800 H.P. per engine. This fact, not before observed, also speaks for the application of turbines for ship propulsion. The section back of the hub is unavoidable, and the axial thrust of the hub requires power as well as the immersed midship section.

All the technical devices cannot change these facts, but we can choose the diameter of the hub as small as possible when designing this.

The frictional losses of the propeller through the water are contained in the constant *C*. They could not be neglected when ascertaining experimentally the value of *C*. It was therefore unnecessary to mention the frictional losses in deducing the theory, and to express these by a separate coefficient.

SOME NEGLECTED ASPECTS OF CYLINDER CONDENSATION.

The following article appeared in series in the "London Engineer."

We propose in this article to deal fully with the question, and set forth as far as possible within reasonable limits facts from which alone useful opinions can be deduced.

The point at issue, tersely stated, is this: Does or does not the multiplication of cylinders reduce what is known as "the missing quantity?" That is to say, will it be less in a compound engine than it is in a single cylinder, and less in a triple-expansion engine than it is in a compound engine? Incidentally it may be added that a reduction in the missing quantity promotes economy. We may begin by calling attention to the fact that no one has yet been able to prove by direct experiment, once for all, that the multiplication of cylinders is by itself able to reduce the missing quantity. The so-called proofs are always based on a theory of condensation, which so far as evidence goes has small basis of fact. What really takes place in an engine cylinder is extremely complex. No one knows enough about it to dogmatize.

Our first step will be to consider what takes place in single-cylinder engines. The most complete and exhaustive experiments ever undertaken with such engines were carried out by Mr. Benjamin Isherwood, Chief of the United States Navy Bureau of Steam Engineering, with the engines and boilers of a number of warships. The results were published in a thick folio volume in the year 1865. What was true of steam engines in that year is true of them now. In the preface in 1865 he wrote: "The diversity of opinion on the subject of the economy to be derived from the use of steam very expan-

sively may be chiefly attributed to our exceedingly incomplete knowledge of the physical qualities of vapors and of the forces of Nature, which has caused mathematicians to apply to steam the laws of a perfect or imaginary gas, totally ignoring its properties as matter, the forces involved, and the phenomena incident to its action; in place of which they have endowed it with purely hypothetical qualities with as much confidence as though they were ascertained facts, and proceeded to examine it as a mere problem for mathematical analysis. But in dealing practically with the subject we must include the effect of these physical properties, and ascertain in what manner they modify the mathematical determination; what are these natural forces and how far they act, and to what extent the results are affected by attending phenomena. We must not substitute fancies and loose slip-shod inferences from them for severe and extensive physical investigations of the question. The immense discrepancy between the theoretical results, inferable from Marriotte's law, and the practical results by direct experiment on the expansion of steam in steam engines proves the utter unworthiness of a metaphysical and mathematical treatment of a physical problem. To complete a refutation of the sophistical nonsense of amateurs and would-be engineers who deem *à priori* argument sounder, as it is certainly cheaper and easier, than laborious experiment, is a lesson which ought never to be lost, and should remain a perpetual warning in the future."

The veteran engineer has lived to see statements which evoked a storm of reprobation tacitly accepted as true. But the problem presented by the steam engine is still very far from being settled. The unprejudiced interested observer who has kept in touch with what has been written and said about cylinder condensation cannot escape the conviction that in this matter, as in most others, disputants have first arrived at conclusions, and then picked out from the multitude just those facts which support their theories, ignoring all others. Nor is this done with any premeditation of dishonesty. It is a policy the result of conviction, due to a lack of the sense of

proportion. It is, of course, quite unscientific, but probably unavoidable.

The first person who saw in cylinder condensation a source of avoidable waste was James Watt. The result of his cogitations was the separate condenser. Instead of putting cold water into the cylinder, he put it into an extension of the cylinder. Following this idea up, he excluded cold air from the interior of the cylinder, and substituted in his "atmospheric" engine an atmosphere of steam for one of air. From thenceforth the idea was driven home in men's minds that the cylinder should be kept as hot as possible; and the circumstance that hot steam would transfer its heat to cooler iron was accepted, not only as a fact, but as a fact of great importance. But how the transfer took place, the rate of transfer, the result of the transfer, or the factors and their bearing on the phenomena, no one troubled his head about. "The missing quantity" had not been named. No one knew that there was a missing quantity. The fact of interest for us just now is that the belief became rooted that cold iron condensed steam at a rate measured by the difference in temperature and by no other factor. Although more recently sounder views have been advanced, and it is understood that cylinder condensation is not a simple but a complex phenomenon, the original faith has never yet been eradicated; and down deep in hearts of most engineers lies the belief that difference of temperature between steam and metal is the ruling factor in settling the amount of cylinder condensation. It is nothing to the purpose that adequate experiments have demonstrated that this idea is probably entirely erroneous. It is still held as an article of faith by the multitude.

Probably the first person who attempted to treat the question scientifically was M. Combes, who, as far back as 1843, published papers in the "Acadèmie des Sciences," on the theory of the steam engine. Writing about the steam jacket in 1845, he stated very clearly the theory of initial condensation and re-evaporation. Thus, "Jackets have not for the main result the maintenance of temperature during expan-

sion; their use consists in the prevention of refrigeration of the cylinder during the exhaust period. "Various other engineers took the subject up. Among the most important investigators was De Freminville, about whom we shall have more to say. In 1851 Le Chatelier, Polonceau and Petiet took part in various discussions. That initial condensation should waste steam, and was affected in amount by the ratio of expansion, was a proposition advanced about 1855 by the late D. K. Clark, and a little later by Benjamin Isherwood. Clark's statement referred to locomotive cylinders, with which alone he concerned himself. Isherwood's had a general application. He stated that no economy could be realized by pushing expansion beyond a certain ratio, because so much steam would be condensed in the cylinder that all that could be gained by expansion and much more was lost. It is a suggestive fact that his views raised a storm of opposition. All the principles of thermodynamics were said to be violated by such a heterodox proposition, the work and theories of French engineers being ignored. So far as we are aware he was the first man who carried out on a large scale an experiment intended to test the true value of expansion. Farey, it is true, was early in the field, but his inquiries took a different turn.

Isherwood's position in the navy of the United States gave him great opportunities, and of these he availed himself. The results of his inquiries fill, as stated above, a large volume. We shall select a single set of typical experiments. They were made with the engines of the wooden war steamer *Mackinaw*, a paddle boat. She was tied up to a wharf. All the feed water was measured with minute accuracy in tanks, and the coal weighed. Indicator diagrams were taken at regular intervals, and each "run" lasted forty-eight hours. The conditions prevented the engines being worked at full power, partly because of the commotion set up in the harbor by the paddle wheels, and partly because the ship did not move. The slow speed of the engine, however, was in this case a distinct advantage, serving as it did to accentuate the phenomena of heat interchange.

The single cylinder was 58 inches diameter and 8 feet 9 inches stroke, inclined, with two double-beat poppet valves about 15 inches in diameter at each end, one for steam, the other for exhaust, worked by Stevens cam gear, giving a very smart cut-off. The engine was fitted with a Sewell's surface condenser. The boilers could supply very much more steam than was wanted. The result was that the steam was delivered free from priming to the engine. The experiments consisted in keeping the number of revolutions as nearly constant as possible, only varying the ratio of expansion. This implied that the mean pressure should be about the same, and this was got by raising the initial pressure with the higher measures of expansion. The results are given in a long table about which, among other things, Isherwood says: "On line 60 will be found the per cent. of the steam evaporated in the boilers which remained condensed in the cylinder at the end of the stroke of the piston by other causes than the production of power. Now, it must be distinctly borne in mind that these quantities do not express the whole condensation which took place in the cylinder. A portion, and probably a large portion of the condensation during the first part of the stroke was re-evaporated during the last part under the continuously lessening pressure caused by the steam after it was cut off; consequently the shorter the steam was cut off the greater would be the re-evaporation, so that the quantities on line 60 do not express correctly proportionally the condensations due to other causes than the development of the power; these condensations being, in fact, much larger proportionally for the higher measure of expansion than shown on line 60. Great, therefore, as a condensation of 37.82 per cent. appears when cutting off at .21, the real condensation must have been far greater. It will be observed that these condensations increase with the measure of expansion with which the steam is used, rising from 7.46 per cent. when it is cut off at 0.70 of the stroke of the piston at the commencement, to 37.82 per cent. when it was cut off at 0.21 of the stroke of the piston." The experiments with the *Mackinaw* were repeated with several other

ships. The interest of the passage which we have quoted lies in the fact that, to all intents and purposes, for the first time it told the engineering world the quantitative story of initial condensation and re-evaporation, as ascertained by a practical experiment carried out on a large scale.

We reproduce "line 60."

	Cut off at			
	0.70	0.56	0.38	0.21
Per cent. of water evaporated in the boiler not accounted for by the indicator.....	7.46	11.43	21.82	37.82

The first impression conveyed by "line 60" is that the cylinder condensed $37.82 - 7.46 = 30.36$ per cent. more steam when steam was cut off at 0.21 of the stroke than when the cut-off took place at 7.46. This is just one of the mistakes which are made most often by writers and speakers dealing with engine economy. It would only have been true if the conditions had been identical, which they were not. All the evaporation figures are given in thousands of pounds. The pounds of water pumped into the boiler when the cut-off was at 0.70 were 523,682; when the cut-off was at 0.21 they were 398,578. Now, 7.46 per cent. of 523,000 pounds is 39,000 pounds, and that was the weight of steam missing with the late cut-off. In like manner, 37.82 per cent. of 398,000 pounds is 150,444 pounds. If we compare the percentages as given in "line 60," it would appear that the condensation was more than five times as great with the cut-off at 0.21 as it was at 0.70; in reality it was less than four-fold.

We may now try to trace out what took place in the cylinder.

To the quantities given in line 60 must be added the percentage required for the performance of work. About this Isherwood wrote: "There is another condensation of steam in the cylinder independent of and additional to that which is effected by the variable temperature of its surfaces, namely, that which is due to the production of work. This condensation, however, differs materially from that already described, in that it does not take place upon, and is not affected by, the

cylinder surfaces. It takes place uniformly throughout the whole mass of steam in the cylinder, and the resulting water, in the form of extremely fine mist or fog, is held in whole or in part suspended by the steam. If any of the water is precipitated on the metallic surfaces it will be re-evaporated during the exhaust stroke of the piston at the expense of the fuel, and no work will be obtained from it."

To this statement exception may be taken. Some additional facts have been obtained since Isherwood wrote, and he has overlooked one of the neglected points. He shows that in the case of the *Mackinaw* with the cut-off at 0.21 the liquefaction reached 7.84 per cent., and cutting off at 0.70 it was 8.72. The total missing quantity, therefore, was in the first case 45.66 per cent., and in the latter 16.18 per cent. During expansion it is commonly assumed, and the indicator seems to prove it, that re-evaporation goes on continually. We shall show further on that this is by no means certain. Are we to assume that the work done by expanding steam does not produce liquefaction, or that liquefaction can proceed *pari passu* with re-evaporation? We have here a point which has never received adequate consideration. Let us assume that it cannot; then the whole of the liquefaction due to the performance of work must take place during the admission period. But many persons, Sir Alexander Kennedy among the number, hold that no work whatever is done in the cylinder during the admission period. It is all done in the boiler. The steam simply constitutes a sort of plug or extension of the piston pushed out by the boiler, and any liquefaction must take place in the boiler—that is to say, if liquefaction is conceivable at all under the circumstances. We see no reason to dispute the soundness of this conclusion. But then we are left on the horns of the dilemma already stated—either no liquefaction or no re-evaporation can occur, or else they must go on together. We shall return presently to the phenomena of re-evaporation. But before doing so it is necessary to consider those of condensation and liquefaction more fully. In unjacketed cylinders, such as that of the en-

gine of the *Mackinaw*, it is well known that on the whole the curve of pressure agrees within a very small percentage with Marriotte's hyperbola. Isherwood writes: "An analysis of a vast number of indicator diagrams from well-conditioned engines of many types and proportions, using steam with varying measures of expansion, and fitted with great diversity of valve gear, shows the total mean pressure to be almost exactly that which is due to Marriotte's law, namely, the pressures are invariably as the spaces, including in the latter the contents of the clearance and steam pumps; and this agreement is not affected by the steam being in the saturated or superheated condition, and with any degree of superheating. In the case of saturated steam the curve falls at first a little too rapidly, and at last not quite rapidly enough. The reverse is the case with superheated steam; when saturated steam is used the pressure at the end of the stroke of the piston is very little greater than is due to Marriotte's law, and when superheated steam is used it is very little less."

The agreement with Marriotte's law has always been a matter of common knowledge, and it has always been regarded as a coincidence. But there must be some reason for this coincidence. With adiabatic expansion even steam gas cannot follow Marriott's law, if work is being done. In the case of saturated steam we must believe that the water provided by initial condensation supplies so much heat that we have isothermal expansion, the whole of the heat used up in the performance of work being thus supplied. We have here a point which has never received the consideration it deserves. The practical fact cannot be disputed. The explanation that it is a mere coincidence repeated thousands of times is wholly unscientific. Nor will it do to say that the curves are not precisely Marriotte's. That they should be nearly Marriotte's at all under so many varying conditions is very remarkable. It is well known that physicists are by no means agreed as to what the curve of steam expanding and doing work ought theoretically to be. Rankine writes:* "From the results of

* "A Manual of the Steam Engine and other Prime Mover."

numerical calculations of the co-ordinates of adiabatic curves for steam it has been deduced by trial that for such pressures as usually occur in the working of steam engines the relations between these co-ordinates is approximately expressed by the following statement: The pressure varies nearly as the reciprocal of the tenth power of the ninth root of the space occupied; that is to say, in symbols $p \propto a^{-\frac{10}{9}}$ nearly." It suffices here to mention this. The reader can pursue the question of the theoretical curve in various treatises on the steam engine. The fact of the so-called coincidence remains, and claims an explanation which so far it has not received.

We now proceed to consider what probably took place in the cylinder of the *Mackinaw*. All the conditions were as favorable as can be conceived to the waste of steam. The pressure was low; the ratio of expansion small; the piston speed only about 110 feet per minute, the revolutions per minute being under 7. The total missing quantity at the end of the stroke is 45.66 per cent. when the cut-off took place at 0.21 and 16.18 per cent. when it took place at 0.70. In the first case 7.84 per cent. was liquefied—by calculation—in the performance of work; in the second case 8.72 per cent.

Let us take the cut-off at 0.21 of the stroke. When the admission valve opened steam entered. Of this 37.82 per cent. was condensed, ostensibly when the steam valve was open. During the earlier phases of expansion a small proportion may have been turned to water, but there is no reason to think that this substantially affects what follows. We have shown already that the actual weight condensed amounted to 150,444 pounds in forty-eight hours, or 3,135 pounds per hour, omitting fractions. The total condensing surface available we cannot give precisely, but it could not have exceeded 140 square feet, in which are included two piston faces, two cylinder covers, and about one-half the cylinder walls. This gives condensation at the rate of, in round numbers, 22 pounds of steam per square foot per hour; the absolute pressure, 53 pounds; the exhaust pressure, 1.84 pounds; the temperatures,

284.6 degrees and 120 degrees; the difference, omitting fractions, 164 degrees.

It will be seen that under these conditions apparently considerably more than one-third of all the steam made by the boilers was condensed by 140 square feet of cast iron. But the surface condenser contained no less than 2,293 square feet. The temperature of the injection water was 44.5 degrees, and that of the steam at the moment the exhaust valve opened 213 degrees; the difference, omitting fractions, was 168 degrees. The cylinder as a condenser was transcendently more efficient than the surface condenser itself. Carefully lagged, always hot, we find, nevertheless, that 140 square feet could do 37 per cent. of the whole work done by 2,293 square feet of refrigerating surface kept constantly wet with cold water. No doubt the condenser was larger than was necessary, because the engine was not working up to its full power. After every allowance is made, however, the difference in cooling efficiency is astonishing.

It may be objected that the conditions prevailing were not the same in the condenser and the cylinder. The point deserves examination. The steam was delivered to the cylinder intermittently; condensation was followed by re-evaporation. But in just the same way, and at just the same rate, steam was delivered to the condenser; the fluctuations of temperature were just the same in it that they were in the cylinder in point of time. The only difference is that there was perhaps no re-evaporation from the metallic surfaces in the condenser. Are we to assume then that in re-evaporation and not in condensation lies the key to the whole problem of initial condensation—that in this re-evaporation lies some subtle power of abstracting heat and cooling down the cylinder metal which has no parallel in a surface condenser? Again, what shall be said of the evaporative efficiency of the cylinder as compared with that of a boiler? Where shall we find a boiler which will evaporate 22 pounds of water per square foot per hour? and that with a temperature of 2,500 degrees at one side of a plate and 300 degrees at the other.

The two boilers of the *Mackinaw* had each 2,500 square feet of heating surface, or 5,000 square feet in all. If they had been as efficient as the cylinder, they would have produced 110,000 pounds of steam per hour; sufficient at 30 pounds per I.H.P. per hour for over 3,600 H.P., or about four times the actual maximum power that could be got out of the engine.

It is not remarkable, we think, that some able engineers have refused to accept such a statement as that made above as true, and have cast about for other ways of accounting for the missing quantity, or have sought explanations of the astonishing frigorific and evaporative powers of cylinder metal.

Among those who have investigated this question, the most recent and most important writers are Messrs. Callendar and Nicholson. They refuse to believe that an adequate interchange of heat can take place between the steam and the cylinder, and attribute the missing quantity mainly to leakage past the sliding valves of an engine. We say sliding, not necessarily slide valve—thus a Corliss valve is a sliding valve, but not a slide valve—such a valve, they say, may be quite tight when at rest, but leaks when in motion. These gentlemen set themselves to measure the range of temperature at various depths within the metal of a cylinder, and satisfied themselves that the surface temperature, instead of following that of the steam, only went through a very small range. Thus with their experimental engine at 70 revolutions per minute, whilst the steam temperatures varied between 335 degrees Fahrenheit and 212 degrees Fahrenheit, a range of 123 degrees, the temperature of the inside surface of the metal only varied 7 degrees. From this it was easy to show that only a small percentage of the missing quantity is due to condensation. Their explanation is, as we have said, that the difference is leakage past the slide valve. Furthermore, they deduced that the rate of condensation of steam, instead of being infinite, is limited. They hold that the rate at which steam condenses on a metallic surface is propor-

tional to the difference in temperature between the steam and the metal, and may be represented by the following equation: B.T.U. exchanged per second per square foot $= 0.74\ (T - \theta)$, where T = temperature of steam and θ = temperature of metal in degrees Fahrenheit.

Messrs. Callendar and Nicholson do not stand alone. Everyone who has considered the subject admits that something remains to be explained.

Unfortunately for the leakage theory, the missing quantity appears to be quite independent of slide valves. The engine of the *Mackinaw* had poppet valves; they are at rest on their seats when closed. It is the easiest thing in the world to make sure that they are practically steamtight, and this Isherwood did.

Some experimenters have satisfied themselves that in compound and triple engines conditions may exist under which the cylinder walls are never dry. It is necessary to call attention to certain obvious facts. No matter what occurs in a cylinder, the occurrences must be cyclical. That is to say, begun and ended in one double stroke of the piston. Whatever the boiler transfers to the cylinder, whether as steam or water, the cylinder will hand on to the condenser. The only difference will be due to the effects of external radiation and conduction and leakage, if any, into the air, and performance of work. This means simply that if steam is condensed during the steam stroke, it must be evaporated during the exhaust. If it is not, then heat would accumulate in the cylinder until condensation ceased, as, in point of fact, happens when an engine is first started. On the other hand, re-evaporation, if in excess of condensation, would cool down the cylinder so much that it could no longer go on. Priming will in effect pass through the engine unaffected. The accumulation of water, therefore, in a cylinder from what is known as initial condensation ought to be physically impossible. Even this appears, however, to require qualification.

The cycle then seems to admit of being stated thus: During admission, part of the steam is condensed, none of it is

liquefied by the performance of work. During the expansion period evaporation goes on simultaneously with liquefaction. During exhaust all the remaining water in the cylinder is evaporated until such time as the exhaust port closes. Then compression begins, and the pressure will continue to rise unless there is water present in sufficient quantity, when liquefaction will begin again, the action then being precisely as though the water absorbed the steam. Then steam is admitted, and the cycle is repeated. It is exceptional, however, to find any water lying in well-made engines properly worked. It is easily drained off.

There are very great difficulties in the way of accepting this account of what goes on in a cylinder. That difficulties exist is proved by the number of hypotheses advanced to explain recognized phenomena.

The first is that the condensation seems, as we have shown above, to be out of all proportion to the ostensible agencies bringing it about, while it is so capricious in its magnitude that it is simply impossible to prepare any formulæ by which it can be calculated. The presence of a spoonful of water in the cylinder will enormously augment the missing quantity. A little oil will diminish it. We can understand that water may in some way promote condensation, but how can it promote the re-evaporation which must also go on? It is too often forgotten that every factor must work both ways. Whatever agency facilitates condensation must in precisely the same degree promote re-evaporation. A jacket will check condensation, and help re-evaporation. It cannot at the same time promote condensation. Conversely, cooling a cylinder outside will promote condensation, but it will certainly not favor re-evaporation. The double or reversible action to which we refer appears to be a point which has not received sufficient consideration.

Then we have liquefaction, which must not be confounded with condensation. Liquefaction is due to the transfer of heat into work. The assumption is that water in the condition of mist results, as stated by Isherwood. To this it may

be answered that it is impossible to prove the truth of the theory by direct experiment. It has long since been proved conclusively that water vapor cannot take the form of mist unless dust is present. How can dust exist inside a cylinder? No one has the smallest idea of the way in which the transformation takes place.

Two views which have obtained some credence may be stated here. The first is that what is called initial condensation is the result of the movement of the steam—the transfer of part of its pressure energy into kinetic energy—that of flow. The collision of the steam with the piston is in effect very similar to its collision with the blades of a turbine—only to a great disadvantage; and that if this will not suffice as an explanation, then that, in some way not understood, the mere impact of steam on an opposing surface will upset its equilibrium and bring about condensation. No proof can be given of this, unless the statement that, no matter how many "separators" steam is passed through, water will be found in them all, can be regarded as a demonstration. The second and far more plausible view is that no liquefaction 'takes place, because the water of condensation in the cylinder can supply all the heat that is necessary. The fact that it is almost impossible to obtain an indicator diagram the curve of which certainly shows the drop due to performance of work may be taken as supporting this theory.

Before we deal with the phenomenon of re-evaporation it is well to consider for a moment what takes place in a compound engine. Theory may be ruled out of court. It would be waste of time to reason as to what ought to happen. We have only to deal with what does happen.

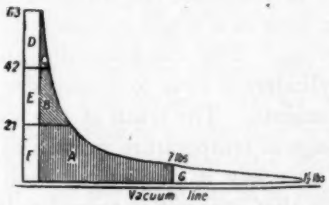
Numbers of experiments have been carried out, both at sea and on land, with compound and triple-expansion engines. The disheartening thing about these trials is their incoherence. No two engines will give the same results, and it has so far passed the wit of man to explain why. Qualitative facts agree always; quantitative facts only by chance. Thus it is always true that, other things being equal, the larger the

range of expansion the more economical will the engine be. But there certainty ends. We cannot trace any invariable relations between the ratio of expansion and the consumption of steam, which can be reduced to weight used per indicated horsepower per hour.

We can take figures anywhere at haphazard to prove this. Compare for the moment the compound engines of the *Colchester*, cited by our correspondent, Mr. Hide, in our impression for May 1st, and that of the *Mackinaw*. The missing quantity in the *Colchester* is 47.3 per cent. against 45.66 per cent. for the engine of the *Mackinaw;* and it must not be forgotten that the *Mackinaw's* was developing only a percentage of her power. Even the *Meteor*, with triple-expansion engines jacketed, lost 24.7 per cent. We have already shown what a well-designed simple engine can do. Here we have a comparison between excellent compound and triple engines and a simple engine working under every conceivable disadvantage. What has the heat trap done here? Major English, R. E., a most careful experimenter, after a prolonged painstaking inquiry, stated years ago that he had completely failed to trace any definite relations between range of cylinder temperature and initial condensation. Elaborate experiments carried out with a number of engines in the United States, which were worked alternately with and without condensers, brought out the remarkable fact that the missing quantity was greater when the exhaust went into the air than when it went into the condenser, although the range of cylinder temperature was, of course, much less.

Those who hold the heat-trap theory as an article of faith very often cling to it because they fail to see why the multiplication of cylinders should promote economy on any other ground. They are so far right in that theoretically the number of stages into which the expansion of a gas is divided has no effect on the ultimate result. The compound engine is, nevertheless, more efficient than the single-cylinder engine, and they attribute the saving in fuel to a diminution in the weight of the missing quantity. We have here one of the ne-

glected points about cylinder condensation which this article is intended to bring into prominence. It is that the introduction of a second cylinder alters the working *régime* radically. The compound engine in daily life is nearly always worked at a higher pressure and ratio of expansion than the single-cylinder engine. In the few cases in which it is not, the introduction of the second cylinder effects no economy. We are dealing now with the steam engine as it is found in mills and ships, not as it is met with in college laboratories. Nothing, perhaps, can better illustrate what we mean than the work done by Macnaught in Lancashire and Yorkshire some sixty or seventy years ago.



Let the accompanying diagram represent the energy in 1 pound of steam, with a pressure of 63 pounds. The total quantity of heat expended in making it, with feed water at 60 degrees, is 1,172 B.T.U. The final condenser pressure is 1½ pounds. Let the engine have a single cylinder working with 21 pounds steam, expanded three times; the terminal pressure is 7 pounds. F represents the full pressure part of the stroke, A the expansion part, F+A represent the work got out of the steam. Now Macnaught found beam engines driving cotton and woolen mills all over Lancashire and Yorkshire working with pressures as low as 21 pounds. In practice it requires no more coal to make steam of 63 pounds than it does to make steam of 21 pounds pressure. Knowing this, Macnaught doubled the boiler pressure to 42 pounds and put in a second cylinder. This was sometimes fixed between the beam center frames and the crank, so that the piston had about half the stroke of the original piston. In other cases

a horizontal or an inclined cylinder was used, a new and larger crank pin being provided. In this way he utilized all that portion of the diagram marked E B, and greatly reduced the fuel bill. If we add a third cylinder working at 63 pounds we utilize D C. We have in effect three steam engines instead of one, and these three require no more steam than F A did. This is the whole essence of compounding in a nutshell. It is the simplest and crudest way of stating the case. The statement is incomplete, but it is not inaccurate. Macnaught achieved a great success. We believe that there are "Macnaughted" engines still running in out-of-the-way corners of Manchester and Lancashire.

It is, of course, obvious that the whole of the work of expansion might be done in a single cylinder; but it would not be good engineering. The compounding and tripling and quadrupling of cylinders is done to reduce stresses and to get better turning moments. The truth of the idea that the reduction in the range of temperature in each cylinder is a factor in producing economy has never yet been demonstrated, while the evidence that it is not is exceedingly strong. Our contention can be best put into the form of a definite proposition.

The multiple-cylinder steam engine of commerce is more economical than the single-cylinder engine of commerce, because it works at a higher pressure and with larger ratios of expansion.

In a sense all this has been a digression. But it was essential to clear the ground, and we may now return to the consideration of the phenomena of condensation.

It is not very easy to get examples of tests of large triple-expansion engines which are not complicated by the use of jackets. We select, however, a triple-expansion rotative pumping engine at Milwaukee, which at one time attracted a great deal of attention because of its high economy. It was put to work about 1894. It was constructed by the E. P. Allis Company from the designs of Mr. Reynolds. The contract stipulated for 18,000,000 gallons per 24 hours, delivered

under a head of 160 feet, the guaranteed duty being 125,000,000 of foot-pounds per 100 pounds of anthracite. This engine was reported upon two or three times.

The most complete trial was carried out by Professor Thurston, and made the subject of a very long and elaborate paper which he read before the American Society of Mechanical Engineers in 1894. The only points of interest for us are the cylinder condensation and re-evaporation. The vertical cylinders are 28 inches + 48 inches + 74 inches × 60 inches. Revolutions per minute, 20; speed of piston, 200 feet per minute; steam pressure at engine, 128 pounds; vacuum, 13.75 pounds. The cylinders are all jacketed, the pistons working in liners. The steam in the first two jackets is at boiler pressure, in the last at 34 pounds. The clearance is very small. There are two intermediate receivers heated by boiler steam. It will be seen that we have here all the conditions most favorable to a reduction in the missing quantity. It is difficult indeed to see why condensation should take place at all; for the jacketing is so complete that it includes the valve chests. Yet "the measurements of the indicator diagrams as originally practiced by Hirn show a progressive increase in the internal condensation of from 12 per cent. in the first and 14 per cent. in the intermediate cylinder to 18 per cent. in the low-pressure cylinder." A combined diagram is given, from which we find that at the moment when the admission valve closed 11.9 per cent. of the steam was missing. The percentage condensed in the jackets is nowhere directly given, but in the course of the discussion which followed the reading of the paper it was said to be 9.25 per cent. apparently for the high-pressure cylinder alone; but this is not certain. Nothing was said of the receiver jackets or those of the other cylinders; but we have no reason to doubt that here in this most elaborately heated engine the missing quantity reached 28 per cent. Professor Thurston deducts a percentage for work done, and goes on:—"It thus appears that the jackets reduced initial condensation, which, in an unjacketed engine of this size and action, would, as

known by experience, be at least 25 per cent., and possibly 30 per cent., to 12 per cent. initially, and to a final 18 per cent.—which would have been 30 or 40—of 20 per cent."—Here we have a man of high reputation, and much experience, conceding with one stroke of the pen that there is nothing in the heat-trap theory, in so far as it means reduced condensation as a result of reduced range of temperature. We may add that his calculations as to the percentage liquefied are wrong, because he has assumed that work was done in the cylinder during the admission period, which, as we have shown in our last impression, is not the fact, the conversion of heat into work occurring not in the cylinder but in the boilers, the true initial condensation—not liquefaction—being about 12 per cent. for a cylinder jacketed with steam at boiler pressure. If space were available we could adduce the story of dozens of tests, all of which go to show that there is no traceable relation between the number of cylinders and the missing quantity, and all pointing to the probability, if not the certainty, that steam is missing not only because of an interchange of heat between steam and cast iron, but for some other reason the nature of which has yet to be settled. We may conclude this line of argument by considering what would take place in theory and in practice in a perfectly non-conducting cylinder.

If heat were not supplied to the steam, that is to say, if adiabatic expansion went on while work was being done, liquefaction would take place. The amount would represent the whole missing quantity. This is thermodynamic theory. There is, however, reason to believe that it does not represent what would really take place. We cannot have a non-conducting cylinder, but we can diminish the conducting power of a cylinder. Thus, for example, the insides of the covers and the piston faces can be coated with lead. The specific heat of cast iron is about 0.13, while that of lead is little over 0.03. The conducting power of lead is as 18 is to 56 for cast iron. It will be seen, therefore, that the heat would have to travel much further into the substance of a lead cylinder, and

that it would be much longer in getting through, than under normal conditions. Consequently a material advantage would be gained by substituting lead for cast iron. Accordingly the experiment has been tried both in this country and the United States, but no advantage appears to have been gained. Wherever we turn we find ourselves confronted with a contradiction of some kind. Thus, although, as we have shown from Isherwood's experiments, condensation and re-evaporation are very large, and although, therefore, it is natural enough to assume that range of temperature must be an important factor, no permanent relation of any kind can be proved to exist between range of temperature in a cylinder and the amount of the missing quantity. So perplexing are the facts that it is not remarkable that many engineers casting about for the solution of the difficulty have suggested that the missing quantity is due to the presence of water in the cylinder; that, in a word, it is water and not cast iron that brings about initial condensation. This view was taken by Clausius and Zeuner, and opposed by Hirn. A very sharp controversy took place in 1887. It does not appear that any particular result followed—no converts were made. Incidentally the question of superheating came in, and it is worth notice that Hirn stated that in his experiments, no matter how highly his steam was superheated, it invariably lost the whole of the superheat the moment it entered the cylinder, a fact which he used to show how instantaneous is the transfer of heat from even gaseous steam to a metal. This supplies another contradiction, for it is well known that great difficulty is incurred in raising the temperature of any gas by passing it through heated iron pipes.

But Hirn and Zeuner and Clausius were by no means the first to see that the effect of water in a cylinder deserved careful consideration. We have already referred to De Freminville, who seems to have been one of the first, if not the first, to consider the effect of water. We take the following statement of his views from a paper on "The Theory of the Steam Jacket," read in 1894 by Professor Thurston:

"Suppose the cylinder to contain a certain quantity of water, and let us see what occurs during a single revolution of the crank, neglecting, for the present, the influence of the metallic interior surface of the cylinder, which we will suppose perfectly inert.

"Let n represent the number of cubic inches of water contained in the cylinder at the end of the stroke.

"T = the temperature of that water, which will evidently be that of the entering steam.

"t' = the temperature of the condenser.

"H = the total heat of steam.

"While the cylinder is in communication with the condenser, the mass of water n, in presence of a vapor of low tension and temperature, commences boiling, and a quantity x is evaporated, leaving the water remaining at the temperature t' of the condenser; this remaining mass being $n-x$.

"Then we have the equation

$$n\,T=(n-x)t'+x\,H;\ x=n\left(\frac{T-t'}{H-t'}\right). \quad . \quad . \quad (1)$$

"There then remains in the cylinder a mass of water $n-x$ at the temperature t', and at that instant steam enters again; coming in contact with the comparatively cold water, it condenses to a certain amount x', making the temperature of the resulting mass $n-x+x'$ equal again to T, and we obtain the following equation of the quantities of heat:

$$(n-x+x')\,T=(n-x)t'+x'H;$$
$$x'=(n-x)\frac{T-t'}{H-T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

But

$$n-x=x\left(\frac{H-T}{T-t'}\right)^{*}$$

$$\therefore\ x'=x\frac{H-T}{T-t'}\cdot\frac{T-t'}{H-T}=x; \quad . \quad . \quad . \quad . \quad (3)$$

* Since from (1) $n(T-t')=x(H-t')$.
adding $-x(T-t')=-x(T-t')$;
$(n-x)(T-t')=x(H-T)$.

consequently, the quantity of steam condensed, on entrance into the cylinder, is equal to the quantity vaporized during the exhaust, and the amount in the cylinder remains unchanged.

"If, in the expression $x=x'$, we make $T=230$ degrees, $t'=122$ degrees, $H=1{,}170$ degrees, we obtain $x=n\times\frac{108}{1{,}058}$ $=0.1\,n$; the vapor condensed at each stroke of piston and thrown into the condenser, without having produced its full useful effect, is equal to one-tenth of the quantity of water in the cylinder."*

De Fremінville computes the amount of this action in an engine of 55-inch diameter of cylinder and of equal stroke, and working at the low pressures and expansions then customary, and finds that a layer of water 0.2 inch in thickness, covering the bottom of the cylinder—the engine had vertical cylinders—having an area of 2,375 square inches, would measure 7 liters (6.5 quarts, 1.63 gallons); one-tenth of this quantity, as above, would be equal in weight to the whole quantity of steam supplied per stroke of engine, and the consumption of steam would thus be made double that of the engine with a non-conducting cylinder, and with no moisture present. He concludes: "It may be seen from this fact how important is the accumulation of water, however slight in quantity, in the interior of the cylinder." Assuming this amount of water, just taken, to be uniformly spread over cylinder head, piston and cylinder proper, it would be but 0.032 inch in thickness over an area of 96.93 square feet; which, as he says, "may be compared to the mist on the window pane." "Such a deposit may occur wherever the temperature of the cylinder is slightly below that of the entering steam; and this is always the case, in regular working, unless special precautions are taken." He then goes on to show that this layer of mist may transfer heat to and from the metal of the cylinder, and thus intensify the effect of cylinder condensation. "This being the situation, when the steam

* "Cours de Machines à Vapeur," Paris, 1862, p. 121.

again enters the cylinder it is condensed, yielding its heat to the metal again until all the cooled mass and the water of condensation are restored to the temperature of the entering steam. This effect would not be produced if the metallic surfaces of the cylinder could be maintained at the temperature of the entering steam." He then goes on to explain how this may be done by a jacket. On this showing any difference in temperature, however small, will produce a deposit of moisture in the cylinder; and if water is the main factor in augmenting the missing quantity, then we see that mere difference in range of temperature may have a very small effect. A trifling difference will do as much mischief as a great one.

It is worth while to consider two factors which are neglected. The first, already twice referred to, is the circumstance that no external work is done during the admission period,* and, as a consequence, that when the admission-port valve closes, the steam is dry so far as liquefaction is concerned. All the conversion of heat into work which takes place inside a cylinder goes on during expansion. We quote the following passages from a communication sent by Mr. Isherwood in 1889 to the Institution of Civil Engineers, being his contribution to a discussion which followed the reading of a paper by the late Mr. Willans, on non-condensing engine trials, because it is in effect a development of the views of Clausius and Zeuner, in that it proposes to explain why water in a cylinder should be so potent for mischief. In effect it means that dry steam will not readily exchange heat with cast iron. Wet steam will. The performance of work makes steam wet. Therefore, condensation is augmented by expansion. It will be seen that, according to De Fremenville and Zeuner, the interchange takes place mainly between steam and water. According to Isherwood, the water only serves to promote the exchange between steam and cast iron: "When steam was

* Mr. Parsons has stated that the exhaust steam from his turbines is dry ("Proceedings of the Institution of Civil Engineers," vol. 96). This seems incompatible with the performance of external work.

used expansively in a cylinder, the cylinder condensation was the aggregate of two entirely distinct quantities; one, the condensation due to the work done by the expanding steam, the other, the condensation due to the metallic surfaces of the cylinder, including those of the piston and steam passages, and to the changes of their temperature. The last quantity, however, was almost wholly caused by and bore a direct relation to the first, without which it could have existed to but a very trifling degree. Hence, of the difference between the weight of steam evaporated in the boiler according to tank measurement, and the weight existing in the cylinder at any point of the expansion portion of the stroke of the piston according to indicator measurement, only the part which remained after deducting the condensation due to the power developed by the expanding steam should be credited to the effect produced by the surfaces of the cylinder and their changes of temperature. Wet steam—that was, steam from every molecule of which some *vis viva* had been taken, or, in other words, steam that had been subjected to partial molecular condensation as opposed to complete mass condensation—had the property of conducting heat in a marked degree, while dry steam, on the contrary, was practically almost destitute of this property. This kind of wet steam could only be produced by using steam expansively, and the more expansively it was used the wetter it was, and the wetter it was the greater was its power of conducting heat. From the foregoing it would be perceived how indispensable, in the analysis of steam-engine performance, was the knowledge of the weight of steam condensed in the cylinder to furnish the heat transmuted into the work of the expanding steam. It was the key to the solution of the problem of steam condensation in working cylinders, no rationale of which could be given by any hypothesis that ignored it. Unexpanded steam was dry and transparent in a working cylinder throughout the whole stroke of the piston, suffering scarcely any condensation, no matter how great the difference between the temperature of the entering steam and the temperature of the

exhaust or back pressure. Expanded steam was wet and opaque, and wet and opaque in proportion the measure of its expansion. The surfaces of the cylinder and their changes of temperature were involved in the problem of cylinder condensation only by means of the wetness of the expanding steam, the finely divided water of condensation forming a connection between those surfaces and the interior of the body or mass of the steam in the cylinder, and enabling this mass in function of its capacity as well as of its external surface to abstract heat from the metal surfaces. This action was entirely different from that of steam not doing work condensed on the surface of a cooler mass of metal. In such a case the steam was dry, and only the layer in direct contact with the surface was affected, the condensation being wholly superficial and proportional to the area of the surface and the difference of the temperatures. But in the case of cylinder condensation when the steam was used expansively, the condensation was not only at the superficies of the mass of steam, but was throughout the entire mass, every molecule of which, so to speak, was simultaneously acted upon by the metallic surfaces of the cylinder. Hence, the enormous quantity of condensation which comparatively small surfaces of cylinder were enabled to effect. If any cylinder not doing work, the exhaust port being kept closed and the piston stationary, was put in communication with the steam room of a boiler, the metal of the cylinder being maintained at the temperature of the exhaust steam, by the external application of cold, the condensation of the steam would be entirely superficial, and the water of condensation would be deposited upon the metallic surfaces, and would be insignificant in quantity compared with that which would be obtained from the same cylinder in operation, using the steam expansively between the same temperatures. This great difference of effect denoted an equally great difference of cause in the two cases, which depended on the fact that in the one case the condensation was of dry steam at the surface of contact, while in the other case

the condensation was of wet steam molecularly throughout the mass."

It may be taken as proved by long extended experience that water in a cylinder is always inimical to economy. How are we to reconcile all this on any theory of metallic heat interchange with Professor Cotterill's assertion that water acts the part of a non-conductor. He calculates that the surface film of moisture does not weigh over 0.01 pound per square foot; that it has a thickness of about 0.02 inch, which gives an obstructive effect, impeding the passage of heat into the iron and out of it equal to that of a film of iron 0.02 inch thick. From this it would appear that water in a cylinder ought to promote economy instead of reducing it.

We have now, we think, shown (1) that so far as evidence is available, the multiplication of cylinders and the reduction of the range of temperature in each has little effect in determining the amount of the missing quantity; (2) that an error is very commonly made in laying down theoretical steam curves for comparison with real indicator diagrams, in that it is taken for granted that work is done during the admission part of the stroke, whereas it is done in the boiler; and (3) that even in practice as good a result can be had from a single cylinder as from a good triple-expansion marine engine.

Although writers on the theory of the steam engine seldom agree with each other, on one point they are united, namely, that water in the cylinder is highly mischievous, precisely how no one can prove. We have shown that the presence of quite an insignificant weight of water may be made to account for the whole missing quantity; but it has to be proved that the small weight of water can be present just at the time it is wanted. One party maintains that evaporation is never complete—that, in other words, at no period during the cycle represented by a single revolution is the cylinder thoroughly dry. The opponents of this view hold that the cylinder must be dried, or else heat would gradually accumulate in the metal.

We have dealt with the theories of condensation; we have now to consider those of evaporation. Let us suppose that in a given engine steam is expanded threefold, that the stroke is 3 feet, and that steam is cut off when the piston has moved 1 foot from the end. Clearance is assumed not to exist. Condensation takes place through one-third of the steam stroke, evaporation through the expansion period, or 2 feet, and throughout the whole of the exhaust stroke, or 3 feet. The time during which condensation can take place is one-sixth of a revolution, and evaporation can take place in five-sixths of a revolution. All the phenomena occur in very minute spaces of time; admission may last less than the tenth of a second; evaporation can then have half a second, and so on. With high-speed engines these times are, of course, still smaller. It is one of the arguments against metallic condensation that there is not sufficient time available for the passage of heat to permit any sensible thickness of the metal to be heated or cooled. Let us further suppose that water has been deposited on the cylinder surfaces; this water is hot. What follows requires careful consideration, because it appears to be quite a neglected factor in the question.

When heated water has the pressure on it reduced, a portion of it *will convert itself* into steam. We have put three words into italics because they state a special truth. The quantity that will resume the condition of steam in this way is readily calculated by well-known formulae, which need not be given here. This re-evaporation would take place in a non-conducting cylinder, but in that case the re-evaporation could not be complete. In a conducting cylinder it can only be complete if the metal gives back to the steam what it received. It is quietly assumed that the evaporating water has some subtle power of extracting heat from the cylinder walls and cooling them down; and we have heard the action supposed to take place explained on the theory of the porous earthen jars used to cool water in the tropics, with which it has nothing to do. If the inside of a cylinder were dry, it could not cool down during the exhaust stroke—at least, to

any appreciable extent—because dry steam is a very bad abstractor of heat. Reduction in the temperature of the cylinder walls can only go on while the film of water in contact with them is colder than they are; and the cooling of the water is brought about by the fact that the temperature of steam, and water in contact with that steam, is settled by the pressure and by nothing else. It is an internal affair, and has nothing whatever to do with the temperature of surroundings. Thus we have only to lower the pressure to reduce the temperature, without introducing any external cold agency whatever. In a word, it is the pressure during exhaust that determines the percentage of re-evaporation. A familiar example, often quoted, is the cryophorus, by which a liquid can be frozen in one glass globe by keeping another several feet away from it, but united by a tube, in a mixture of snow and salt. Furthermore, if we have hot water under pressure and suddenly release that pressure, a percentage of the water—say, 20 per cent.—will at once flash explosively into steam. It is in this way that the violence of boiler explosions is explained. Now, although the mysterious change from the liquid to the fluid condition can take place with the rapidity of a lightning flash, it by no means follows that heat can be transferred from iron to water with the same rapidity. It would be fair, then, to suppose that the moment an exhaust port opened a percentage of the water present would be converted into steam by what may be termed the automatic action of heated water when relieved by pressure, while the remainder would dry off gradually as it obtained heat from the metal. It is somewhat disconcerting, therefore, to find that the whole of the water disappears in a fraction of time too small to be noted even in engines running at comparatively slow speeds.

Mr. Donkin, some years before his death, made many attempts to learn by actual inspection what goes on inside a steam engine. To this end he constructed small engines with glass cylinders, but the cylinders cracked and could not be used without serious risk, in spite of wire guards. Then he

fitted to the compound beam engine in his Bermondsey works a "steam revealer," a species of clear-glass bottle which could be put in communication with the clearance space at the top of the cylinder. The phenomena have often been described. When the steam valve opened water was deposited as dew on the glass. When the valve closed the deposit ceased. When the exhaust valve opened the glass cleared instantly. If a transfer of heat took place from the glass to the water it must have been at a rate which is not contemplated by such calculations as those of Dr. Mellanby, given in his paper on steam jacketing, read in 1905 before the Institution of Mechanical Engineers at Liège. His calculations are based on the rate of transfer per minute. Mr. Royds has very properly pointed out in our correspondence columns that this may be very misleading. The right method of reckoning is to take the actual time during each stroke that condensation can go on.

The letter from Mr. Royds which appears in our correspondence columns contains nothing to which we take exception. He believes that valve leakage to a considerable extent takes place in all engines under all circumstances. He is not without followers. But he will do well to remember that by far the greater number of engineers refuse to admit that any appreciable leakage in first-class engines goes on. On this point we would refer him to the "Transactions" of the Institution of Mechanical Engineers for 1905, Vol. II, where he will find the discussion which followed on the first report of the Steam Engine Research Committee. According to Messrs. Nicolson and Callendar, the valves pass water, not steam, and the steam seems to be condensed by flowing through minute leaks. But no satisfactory attempt has been made by any one to show what becomes of the "latent" heat of the condensed steam in this case. There is the further difficulty that in compound and triple-expansion engines the percentage of the missing quantity fluctuates, sometimes being considerably less in the intermediate than it is in the high-pressure cylinder. It is not easy to see how the leakage water handed on to this second cylinder should be converted

into steam in it. We have not quoted "Mr. Isherwood's old-time statements as evidence against valve leakage," but simply to show how the action of water may be explained on one theory.

It will be seen, then, that if the cylinder is dry when the admission port opens, initial condensation cannot be due to water. But in that case the transfer of heat from steam to cast iron must take place at a rate which sets calculations at defiance. Again, in order that the cylinder may be dry, evaporation must be caused by a scarcely less rapid transfer of heat from the metal to the water. All this goes, of course, to support the views of Clausius and Zeuner. Indeed, if we suppose that metal and steam and water cannot interchange heat to all intents and purposes instantaneously, we are compelled to postulate either that water lies permanently in the cylinder, the surfaces being always wet, or else that there is some other agency at work to account for the missing quantity, the nature of which agency has yet to be ascertained. A plausible way of reconciling facts is to take it for granted that boiler steam is always wet. But this is very difficult to prove. Another explanation has already been referred to; it is that the first steam which enters "loses its life;" not because it is cooled down, but dynamically from shock, and as the effect of loss of energy by collision with inelastic walls.* In this way water is produced sufficient in quantity to account for further condensation. It is right to add here that in Donkin's experiments a small quantity of water always remained in the bottles. This seems to be due to loss of heat by external radiation and conduction. Professor Unwin has given a very ingenious explanation of the phenomena.

Whenever steam is condensed it is said to surrender its lat-

* One of the patent difficulties about the molecular-battery-pressure theory of gases is that it presupposes perfectly elastic molecules, which can collide not only with each other, but with the surface of the vessel on whose parts they exert pressure, without loss of energy. Were it otherwise, the pressure in a gas cylinder must gradually diminish unless the gas received heat continuously. But the metal walls of a compressed gas cylinder are not perfectly elastic. Yet the pressure does not fall. To get over the difficulty Sir Oliver Lodge has framed the hypothesis that the gas molecules never come in contact with the metal at all, but with a lining of the ether, which being perfectly elastic there is no loss of energy.

ent heat. Now, this is a very misleading statement unless it is used with caution. There is no such thing as latent heat. All the heat in steam is expressed by its temperature measured by its weight. The "latent heat" represents so much work done in making steam out of water, and this can be caused to reappear either as work of another kind or as "sensible" heat. In a steam cylinder a portion does work on the piston and a portion reappears as heat. The first accounts for liquefaction, the second goes to the condenser. But of the precise nature of the processes going on or the phenomena involved nothing is known with any certainty; nor will anything be known until an inquiry into the nature of steam, the method of its production, and the way in which it parts with its dynamic energy has been carried out.

Reference has been made more than once in our columns to the curious theory of C. Wye Williams, which is in effect that water cannot be heated at all. To accept this view it is necessary to believe that water molecules have no temperature. When heat is applied it is converted into kinetic energy, and steam is produced which is immediately diffused through the mass of water, just as CO_2 is in "soda water," and it is this steam which affects the thermometer. According to this theory the water lying in a cylinder would simply absorb steam at one pressure, as a sponge does water, and give it out again at another pressure. This is precisely what appears to take place in engine cylinders during the compression stroke if much water is present. When a bubble of steam is made to pass down one leg of a glass U tube it will become smaller and smaller until it may quite disappear. As it passes the bend and begins to rise again on the other side it will again form, growing larger and larger until it reaches the surface. The difference of pressure at the top and bottom of the tube is only a few ounces. What becomes of the latent heat when the bubble vanishes? The absorption theory was surrounded with difficulties when Williams first broached it. But many of these disappear in the light of recent researches into the

nature of matter. The position of the theory of the steam engine may best be set forth by an allegory.

Once upon a time a king decided that he would have a pillar erected to his honor, and he called upon all his subjects to contribute stones and gold and iron and all things needful. And his subjects elected certain wise men and entrusted them with the work, and each of these men studied the subject and made calculations, and used up the whole Greek alphabet and much more; and after a time they met and proceeded to build the pillar, and lo! what each man contributed was very good, and nicely designed, and in many ways quite clever. But when they came to put the parts together, behold! none of them would fit, or if, peradventure, two did, they were left out in the cold by all the others, and to this day the pillar remains unbuilt.

On the 13th of March, 1888—that is, a little more than twenty years ago—Mr. P. W. Willans read a paper before the Institution of Civil Engineers on "Economy Trials of a Non-condensing Engine, Simple, Compound and Triple." No more noteworthy paper has ever been read before an engineering society. It occupies, with the memorable discussion which followed it, 196 closely-printed octavo pages of the "Transactions." The trials were remarkable for their completeness and the scientific accuracy with which they were carried out. But the discussion which followed was even more remarkable, because nearly every eminent engineer, professor of engineering or physicist at home or abroad took part in it. We have already quoted from its pages. The prominent feature of the whole was that no two of the speakers agreed, save on a few general principles. No sooner was one proposition advanced as a matter of fact than it was contradicted by another bit of fact from the lips of a different speaker. It was not as though there were two theories—two sets of men in opposition. There were almost as many theories as speakers; and all, be it remembered, first-class men—men of experience, men who advanced nothing that they had not satisfied themselves was true. The whole literature of the subject

may be said to be summed up in this little volume, and in the last twenty years we have made not the smallest advance towards formulating a true theory of the steam engine as an explanation of what goes on inside it. The discussion on Willans' paper represents all that is known today.

The permanent practical present-day fact is that in superheating and the reduction of clearance space lies the source of economy. All improvements in the reciprocating engine must have for their object the exclusion of water from the cylinder and the reduction of clearance space. There is no valid reason why this last should not be as small as it is in ammonia compressors. It has already been brought down to 1½ per cent. It does not appear that any advantage worth the extra first cost is to be had from excessive pressures or temperatures, and it seems likely that the best commercial results can be had from mill or electric generating engines, with 140 pounds pressure in the cylinder at cut-off, and 8 pounds when the exhaust opens, and a superheat of 150 degrees at the engine. The engine to be a cross-compound. Something between 9 pounds and 10 pounds of steam per indicated horsepower may be expected from a thoroughly well made, well designed engine working within these limits. By carrying 220 pounds at sea, and using quadruple engines, a result equivalent to about 11 pounds or 12 pounds of steam per I.H.P. may be had. But the same result might be had with much lower pressures in a triple engine with a superheat of 100 degrees Fahrenheit at the engine and efficient drainage for the receivers.

We have endeavored in writing this article to be impartial. Our object has been to direct attention to some of the many facts and expressions of opinion which are neglected in the present day by those who write about the steam engine. We have advanced no theory of our own ; we have have done no more than suggest. But, judging from the remarkable things in the way of condensation and liquefaction that apparently go on in cylinders, it seems that a field of inquiry remains unexplored which may contain matters of value. The certainty

is that if only steam was a more stable fluid than it is, it would be in like proportion more economically efficient; and it is in the art of imparting this stability that the promise and potency of steam lies, and not in the mere raising of boiler pressure, or the multiplication of cylinders.

S. S. *TENYO MARU.*

TURBINE MACHINERY AND OIL-BURNING BOILERS.

REPORT BY ENSIGN W. S. ANDERSON, U. S. N.

While taking passage, June 11, 1908, from Kobe to Yokkaichi, Japan, on the S. S. *Tenyo Maru*, I was allowed to inspect the turbine propelling machinery and oil-burning boilers of that vessel.

The following information was kindly given to me by the Chief Engineer of the *Tenyo Maru*, Mr. A. H. Seaver, an American, from the State of New York. There are nine other engineer officers, all Japanese.

The *Tenyo Maru* is a triple-screw turbine steamer belonging to the Toyo Kisen Kaisha (Oriental Steamship Co.). Displacement, 21,000 tons, gross 14,000, net 7,265. It was completed in 1908 by the Mitsu Bishi Dock Yard and Engine Works, Nagasaki, Japan.

The turbines, of the Parsons' type, were built by the Parsons Marine Steam Turbine Co., Wallsend, England, in 1907.

There are three 12-inch hollow-steel shafts, each carrying one propeller; pitch, 105 inches; diameter, 115 inches. The central shaft carries one turbine, the high-pressure one. Each wing shaft carries two turbines, one to go ahead, one to go astern, both in the same casing, with a common exhaust. Just outboard and slightly above each wing turbine casing is a surface condenser of the ordinary type, but having also an augmenter.

Steam can be admitted to the central high-pressure turbine, to both the wing ahead turbines and to both the wing astern turbines. Ordinarily, when going ahead, high pres-

sure steam is admitted to the central high-pressure turbine only, passing from that to the wing ahead turbines and from each of those through that wing exhaust to its condenser. In case it is desired to stop, shutting off the steam from this central turbine by one valve stops the machinery. In case it is desired then to maneuver, high-pressure steam may be admitted directly to either the ahead or astern turbines on either or both sides. As the wing turbines on one side have a common exhaust, no exhaust valves have to be fitted and only steam valves need be manipulated.

There is an emergency governor of the ordinary centrifugal type which acts when 330 revolutions are exceeded. A normal number of revolutions for about 20 knots speed is about 303. The normal number of revolutions astern at full speed is probably about 200.

The oil for lubricating the turbines is used over and over again, being circulated by a pump under a pressure of ten pounds, the oil passing through a water cooler.

The steam pressure is balanced by the upper part of the thrust bearing; the propeller thrust by the lower part of the thrust bearing; therefore, if in equilibrium, there should be no thrust on the thrust bearing.

Longitudinal adjustment is provided for by a micrometer gauge in contact with a dummy piston, and also the longitudinal adjustment may be obtained by calipers at a point on the turbine shaft.

Ordinary steam packing is employed.

The blades are of special bronze.

The principal overhauling necessary at the end of a run is to overhaul the bearings.

The total I.H.P. which can be developed ahead is about 20,000; astern, about 7,000. This is measured by means of a torsion meter.

The condenser vacuum is increased by having an augmenter assisting the air pump. Additional suction is obtained by the use of a jet of steam, upon the principle of an

injector in a pipe, connected at one end to the bottom of the condenser and at the other end to the air-pump suction.

While under way at a speed of about 16 knots, the engine-room indicators were read as follows: Revolutions, port, 240; starboard, 240; central, 240. Boiler steam, 165 pounds. High-pressure steam, 90 pounds. Port low-pressure steam, 12 pounds. Starboard low-pressure steam, 12 pounds. Port astern, 27-inch vacuum. Starboard astern, 26.5-inch vacuum. Port condenser, 26.7-inch vacuum. Starboard condenser, 26.8-inch vacuum. Feed-water heater, 2 pounds less than atmospheric. It is thus seen that boiler steam is wire drawn considerably, dropping from 165 to 90. The wing astern turbines should, of course, show the same amount of vacuum as the condensers. It is possible to use as much as 160 pounds steam pressure at the high-pressure turbine.

The engine room is roomy and the temperature not excessive. There was no vibration worth mentioning.

An excellent book on this turbine is the following: "The Marine Steam Turbine," written by J. W. Sothern, published by Whittaker & Co., Paternoster's Square, London, in 1906, price 2*s.* 6*d.* net.

Boilers.—The *Tenyo Maru* has thirteen single-ended oil-burning Scotch boilers. Nine were in use when going at 16 knots speed. They were completed in 1908 by the Mitsu Bishi Dock Yard and Engine Works, Nagasaki. The oil-burning attachment was built by Lassöe-Lovekin (Oil-Burning Co.?), Brooklyn, N. Y., in 1907.

The front end of the furnaces, the lower third of the furnaces and the back end of the combustion chamber are lined with fire brick. Each boiler has four furnaces and each furnace has one oil-burning nozzle. Residuals is burned, at present Borneo oil being used. 1.2 pounds of oil is burned per I.H.P. There were seven men on watch as a firing force, effecting a considerable reduction in this force over that necessary in case coal had been used. This vessel carries 3,500 tons of oil and no coal. The tanks are open to the atmosphere. Everything is kept clean about the tanks and there

are facilities for sucking out gases from the surface of the tanks. The oil burned is heated to a temperature of 140 degrees Fahrenheit and is under a pressure of 20 pounds per square inch. Compressed air, under a pressure of 3½ inches of mercury, enters the nozzle with it. One nozzle consumes 400 pounds of oil per hour. One pound of oil converts 14 or 15 pounds of water into steam from and at 212 degrees Fahrenheit.

As the relative heating efficiencies of coal and oil is about as 4 to 7, and as oil can be bought more cheaply than coal, and also requires a smaller firing force, there is a very considerable economy in the use of the oil. Some figures on the cost of petroleum fuel can be obtained from a pamphlet entitled "Petroleum in California," published by the California State Mining Bureau, Ferry Building, San Francisco, California.

One point about marine turbines which should appeal to the naval constructor is the fact that they do not require the head room that reciprocating engines require, and they could, therefore, be gotten under a lower protective deck and with less congestion of machinery in the engine room. Also, in case of ramming or sudden stopping there are no exposed moving parts into which men or spare parts might be projected.

SUBMARINE MINES AND MINING.

By Major Richmond P. Davis, Coast Artillery Corps, U. S. Army.

The terms "submarine mine" and "torpedo" have been used somewhat loosely; they should have the significance given below.

A submarine mine consists of a case containing a charge of explosive and appliances for firing it, fixed in position beneath the surface of the water. A torpedo consists of a case containing a charge of explosive and appliances for firing it and propelling it through the water. The submarine mine is thus purely defensive in character while the torpedo is defensive-offensive. However, the records show much more damage to vessels by mines than by torpedoes.

Mines may be classified as mechanical and electrical, depending upon the method of firing; as ground and buoyant, depending upon whether the case is placed on the bottom or held in position near the surface of the water.

Mechanical mines have all the firing apparatus in the case, and when planted are dangerous to friend and foe, as they will fire when struck by a passing vessel. Once planted there is no indication of their whereabouts or condition, therefore their employment is a question of expediency. That they are wonderfully effective in destroying both friend and enemy is shown by the experience of the Russians in the late Russo-Japanese War.

The favored method used for firing mechanical mines are as follows:

1. A glass tube containing sulphuric acid is placed so that it will be broken by the jar when the mine is struck and

allow the acid to come in contact with potassium chlorate; the heat of reaction will cause the explosion of the charge.

2. A pistol and ball are arranged so that the ball will be thrown against the pistol trigger when the mine is struck and thus cause a shot to be fired into the charge, exploding it.

Mechanical mines may be planted very readily and are very cheap, hence their extensive use. The operation is very dangerous, and a Russian mine planter was blown up at the very beginning of the late war in the East by one of her own mines. The dangers incident to planting, to raising these mines after the war is over, the chance of shifting from the position at which planted, of breaking from their moorings and drifting in the path of neutral vessels or to the high seas, make their use very precarious. Laws should be enacted to prevent their indiscriminate use and thus eliminate the dangers to neutral shipping which may be passing through the theater of war operations.

Electrical mines may be classified either as controllable or non-controllable. Those of the former class may be rendered dangerous or inert at the will of an operator on the shore, and may be tested at any time. The defense is thus familiar at all times with the whereabouts and condition of every mine, and the mine field is a known and reliable element of the defense. The non-controllable electrical mines are fired by means of a firing battery which is thrown overboard after the planting vessel has anchored the mine and drawn off to a safe distance. After the battery is thrown overboard this electrical mine is to all intents and purposes a mechanical one, and its use has resulted solely from the danger incident to planting the latter. This mine has a further advantage in that it would be rendered inert in all probability if torn from its moorings.

Metallic-mine cases are used universally and are spherical, cylindro-spherical and conical in shape. The first type is the best, as it exposes the same surface to the tide currents, whatever be the submergence, gives great buoyancy and is easy to handle. The case may be made in a single piece, the hemi-

spheres being pressed and then welded along the equator; or it may be in two pieces and prepared for service by placing a rubber gasket between the hemispheres and clamping on the equatorial lip. The former is the more reliable type, but the latter is necessary when the mine cases have to be kept aboard ship and the storage space reduced, therefore, to a minimum. If the case is in a single piece, as shown in Fig. 1, the charge and firing mechanism are inserted through a hole called the loading hole, and this is closed by means of a plug through which the electric conductor passes to convey the current to the fuse; stuffing boxes are provided to prevent leakage. The explosive must be made up of a number of small packages which may be inserted through a hole not more than five or six inches in diameter.

With the type consisting of two hemispheres clamped together when the mine is to be planted, the charge is assembled in a single piece and placed in one of the hemispheres and the other then clamped on. As in the preceding instance, the electric conductor for the firing battery must come out through a stuffing box. Experience shows that the mine cases in a single piece can be kept down with much greater certainty against leakage than those composed of two pieces.

The mine should be moored so that it is just far enough beneath the surface at low water not to be seen. The tidal current has a depressing effect, hence for harbors with swift tides great buoyancy is necessary. Experiment has been made to prevent this extra submergence by an attachment against which the current would work and develop a force contrary to that of the current against the mine case proper. However, the best method is to obtain greater buoyancy by increasing the size of the case as shown in Fig. 2. Experiments have been made with the loading hole and mooring attachment on the side of this modified case, but the tide upended the mine and increased the submergence over that for the normal method of anchoring with the mooring rope at the end.

In case of emergency very useful mine cases can be improvised from good, strong kegs.

An effective submarine mine must have within it a piece of apparatus which will close the electric circuit as soon as a vessel strikes it. This piece of apparatus is called the circuit-closer, and there are many forms of it. In some cases there is a suspended cone with a good deal of inertia, to which the electric wire is connected. A spring attached to the case is placed so that when the case is struck by a passing vessel the cone will touch the spring. The conductor leading into the mine is grounded by this operation, and if the other pole of the battery is grounded also the electric circuit will be completed when a mine is struck and a current will pass through the fuses. The fuses used are of the ordinary type, consisting of a short piece of platinum wire surrounded by mercury fulminate. The wire is heated by the passage of a current and the detonator causes the explosion of the entire charge. To ensure explosion two fuses in multiple should be used.

If the mine be of the controllable type it may be arranged so that the current which passes through the circuit upon the striking of a mine by a vessel will cause an alarm in the operating room on shore but will not fire the fuse. This effect may be accomplished in many ways—these constitute the principal secrets of the mine systems in use.

The explosives which have been used in submarine mines are black powder, dynamite and wet guncotton; of these the last two are the only important ones. General Henry L. Abbot, of the United States Engineer Corps, performed some very valuable experiments in the early 70's with a view to determining the best explosive for use in submarine mines. In this connection he enunciated the following requirements:

1st. The greatest possible effect when fired under water in such envelopes as are suitable for submarine mines.

2d. Retention of normal strength under the conditions incident to the service, to wit: lapse of time, alternate freezing and thawing, occasional wetting and even saturation with water.

3d. Convenience in loading—involving safety in transportation and handling with ordinary roughness—a form which

admits of ready insertion in a hole small enough to be rendered watertight, and a high density in small bulk.

These three conditions are as true today as thirty years ago, but modern requirements compel a fourth—the explosive for submarine defense of our forts must be stored at the forts and ready for instant use.

There should be added also the following: A mine must not be dangerous if torn from its moorings. The peril in the Eastern Seas due to vagrant mines used in the Russo-Japanese war is well known to the navigators of those waters, and on many occasions innocent travelers have been exposed to drifting mines, and merchantmen have been destroyed more than a year after the termination of the war.

The picric-acid high explosives have received much attention for warlike purposes within the last decade, but they are not suitable for submarine mining. The explosive used by the United States in the Spanish war was dynamite of various grades. According to General Abbot's experiments this explosive (seventy-five per cent.) is stronger than wet guncotton in the proportion of seventeen to fifteen, and one hundred pounds of it has a destructive radius for battleships of about sixteen feet. In some recent experiments with another high explosive it was found that sixty-three pounds would produce serious injury at a distance of twenty feet, while one hundred and thirty pounds at a distance of fifteen feet disrupted entirely the double bottom of a target made to represent a section of a battleship. The utmost care must be used in loading submarine mines with dynamite, and it should be put into the mine case in ordinary packages containing about five pounds.

The priming charge to be placed around the fuses should be of loose dynamite, one pound in weight, carefully packed in a small bag (cotton or linen), the fuses inserted in holes made with a soft pine stick, and the whole tied around the choke with a piece of string. On account of the severe headaches which result from handling this explosive it is important that the operators should use rubber gloves. Special attention must be paid to the screw threads of the loading

hole. Before the closing plug is screwed in these threads should be scrubbed carefully with a soft brush.

Sixty per cent. dynamite is the strongest commercial product at present and should be used if this explosive is employed for submarine mining purposes.

The greatest danger to be apprehended in using dynamite is the leakage of the nitroglycerine. This is certain to occur if the least bit of water has access to the explosive. Very serious accidents have resulted from the explosion of a film of nitroglycerine on the inside of a mine case after the entire charge had been removed. If it is necessary to remove the closing plug from a mine case loaded with dynamite some mechanical arrangement should be rigged up so that the plug may be unscrewed from a distance.

Wet guncotton is an ideal explosive for mines and is used universally in modern torpedoes. The priming charge is dry guncotton, and to be certain of detonation arrangements must be made so that the primer will be separated from the charge by a non-metallic substance. In some experiments made at the School of Submarine Defense to determine the limiting distance at which dry guncotton will explode wet, it was discovered that wire mesh of copper between the dry and the wet would prevent detonation, whereas detonation would occur in every instance with a half an inch of non-electrical conducting material such as hard rubber or wood separating the primer and charge. It was found in the course of these experiments that the character of the material separating the primer and the charge was much more important than the dryness of the primer, and that wet guncotton with twenty-five per cent. of moisture could be fired with a primer which was itself far from dry. The strength of wet guncotton is to that of dynamite in about the proportion of fifteen to seventeen, and one hundred pounds of it can be relied upon to damage a ship at a distance of fourteen feet.

Some recent experiments have been made with a view to determining the character of a subaqueous explosion and the results have been very interesting. These experiments were

made with tri-nitro-toluol. $C_6H_4(NO_2)_3$. They demonstrated that there are two phases of a subaqueous explosion—percussive, in the nature of wave motion, and pressure; the former acts first and extends to a great distance, the latter exerts pressure due to a displacement of the material, but does not extend a very great distance from the mine. The former depends upon the rate of detonation and amount of gas; the latter upon the volume of gases, temperature of explosion and density of charging. The pressure effect comes after the percussive, and to the latter the destructive effects are largely attributable. The experiments showed that the percussive effect of tri-nitro-toluol is twenty-seven per cent. greater and pressure effect seven and one-half per cent. greater than those of wet guncotton, bulk for bulk. Furthermore, tri-nitro-toluol is very insensitive to shock, may be stored indefinitely with absolute safety, is not dangerous to handle, and may be used in loading with great facility. It is not affected by water, may be melted and cast into any shape; it may be fired by the detonation of a loose crystalline form of the same material. It will be seen from the foregoing that this explosive fulfils to a high degree the necessary requirements for submarine mining.

It has been noticed often by observers at some distance from an exploding mine that the fish jump from the water before any sound is heard or eruption noticed. The arrival of the percussive wave is very noticeable on a large vessel several hundred yards from the explosion. In one instance a condenser pipe was sheared off when a mine planter was about three hundred yards from the exploding mine, and the firemen often are very uncomfortable on mine planters, even though the vessel may be more than a thousand yards away when the mine is fired. In observing the water just above an exploding mine a mound will be seen to form and then a column shoot up to many feet, the height depending upon the submergence; a light target above the mine is projected upward ahead of the column.

The mine case may be fixed in position by a cast-iron

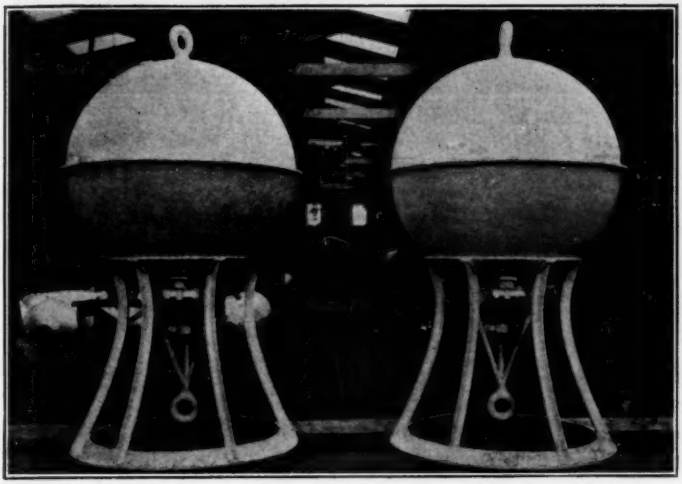


Fig. 1.—Two 32-Inch Mine Cases Supported on Stands. The Loading Hole is Covered by the Cap to which the Mooring Rope is Attached.

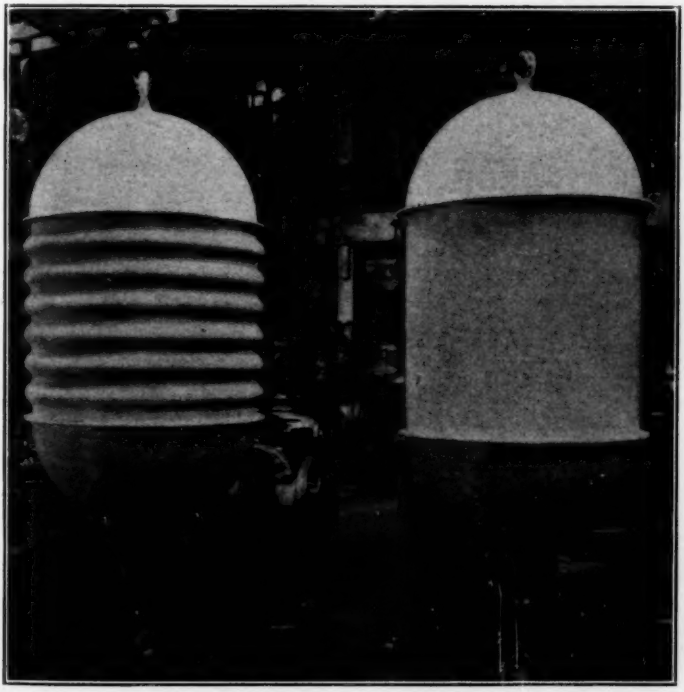


Fig. 2.—Mine Cases for Swift Currents. The Corrugated Cylinder is the Latest Type and this Construction Has Diminished the Weight Very Materially.

Fig. 3.—A Typical Mine Explosion.
The Mine in This Case was Moored so that it was Very Near the Surface.

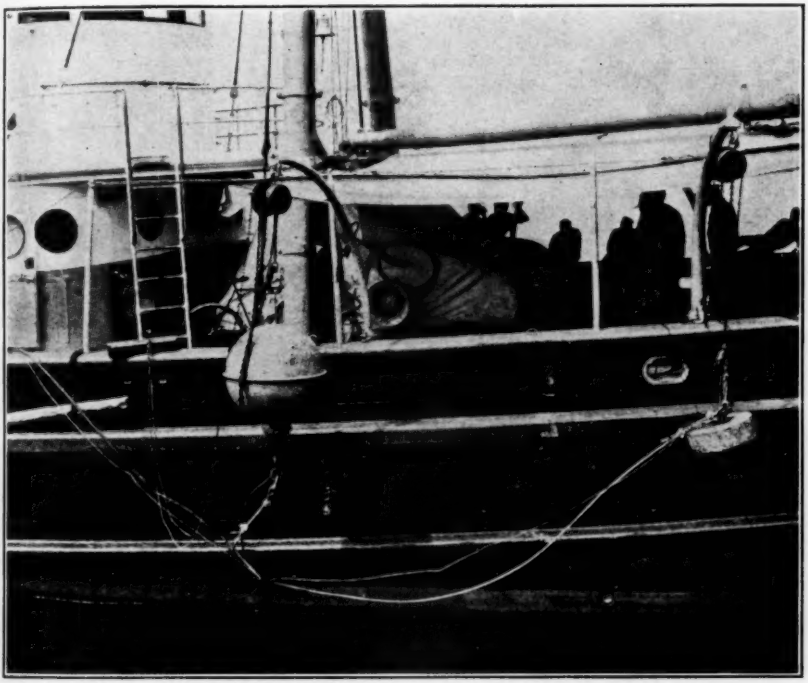


Fig. 4.—This Shows a Mine Ready For Planting.
The Cable Runs Aft where it is Coiled as a Figure Eight.

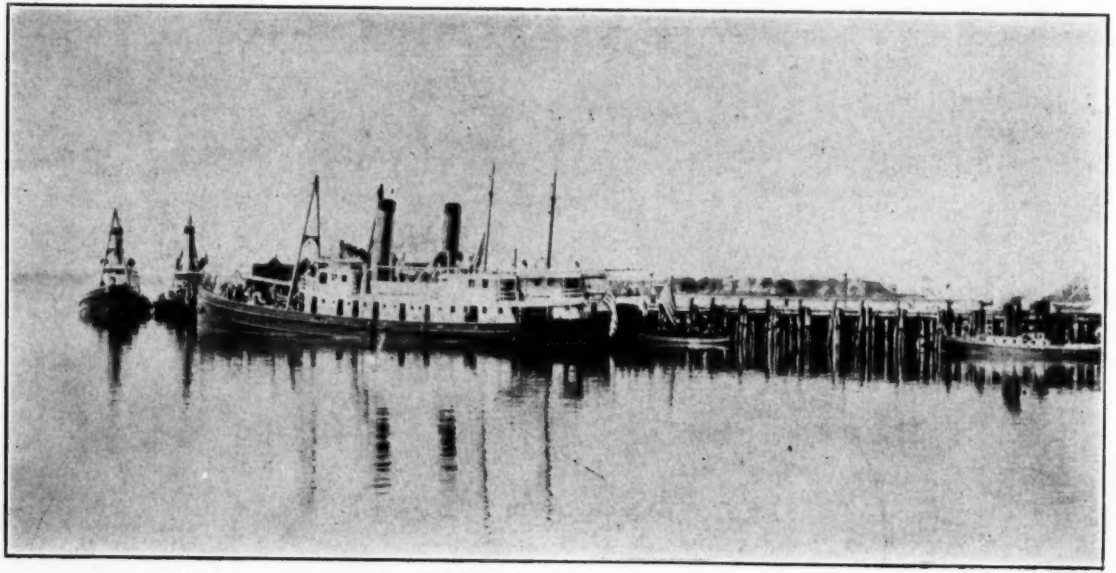


Fig. 6.—A Fleet of Mine Planters.

anchor cylindrical in shape and of a weight depending upon the buoyancy of the mine and the swiftness of the current. In this case it is necessary to take soundings and plant the mines very approximately at the places for which soundings were taken. This is very difficult in water more than one hundred feet deep, and has given rise to experiments to develop an anchor which will enable the mine to be moored at the proper depth below the surface without the necessity of taking soundings. All governments have an anchor of this nature, and most of them consist of a semi-ellipsoidal cast-iron case inclosing a drum upon which the mooring rope is wound. The drum is allowed to revolve during the descent of the anchor until it reaches a certain distance from the bottom when the revolution ceases and the mine is pulled down upon further descent of the anchor. These operations are brought about by having on the drum a ratchet into which a dog is pushed by means of a spring. If the tension of the spring is overcome the dog is pulled out of the ratchet and the drum revolves. A heavy lead weight is attached to this dog so as to overcome the tension of the spring and is of such a shape that its tendency to sink is greater than that of the anchor. The dog is thus pulled out of the ratchet and the drum revolves until this weight hits the bottom. As soon as the weight touches bottom the dog engages and stops the revolution of the drum. The weight is set at a distance from the anchor depending upon the depth at which it is desired to moor the mine case, the state of the tide and character of bottom. Effort has been made from time to time to produce an anchor which would accomplish the purpose noted above and also cause the anchoring rope to be lengthened and shortened as the tide rises and falls, thus producing a constant submergence for the mine case. This has not been accomplished successfully, and I think it is safe to predict that it will not be in the near future. Mines may be planted with facility in any depth of water up to two hundred feet.

The interval between mines may be fifty feet without danger of damage from a neighboring mine when the charge is

one hundred pounds of high explosive. This small interval is not necessary, as the beam of a battleship is more than seventy-five feet and the destructive radius of the explosive about fifteen feet. From these figures the maximum interval at which mines should be planted to close a channel with certainty is about one hundred feet. The mines are planted in groups of any desired number. From each mine a single-conductor cable runs back to a box called the distribution box. From the distribution box a multiple cable runs to the shore, and in the box electrical connection is made between the single conductor and a conductor of the multiple cable. The best system of mining is that where only one single-conductor cable is joined to each conductor of the multiple cable, for in this case each mine is independent of the others. However, if it is considered desirable, more than one mine may be joined to a single conductor of the multiple cable and in this way the mines operated from one multiple cable made to cover a much greater width of channel; independent action and control are sacrificed, but the expedient may be necessary in case of a shortage of multiple cable. From the distribution box the cable runs to the operating room of the fortification, and through a hut when this room is removed some distance from the shoreline. The hut should not be located immediately at the water's edge.

In order to permit friendly vessels to pass through a mine field without danger to the vessels or the mines, the different lines should not extend all the way across the channel, but should overlap, leaving a distance of five hundred or more yards through which a ship may pass when running on specially-located range points known only to certain pilots who would be used to take all vessels in and out of the harbor. In case of extremity these openings could be closed up quickly from the reserve supply of mines for the fort.

The operating room on shore contains all necessary electrical apparatus and appliances for automatic testing and firing of the mines or performing the same functions at the will of the operator. The methods employed for this pur-

pose are also secrets of the different mine systems. An efficient mine system requires that the testing should be simple and firing certain.

Mine planting is a very interesting operation, and the hazards involved with certain classes require the utmost precaution and alertness. Something new is developing always, and some loaded mines should be used constantly to prevent the apathy which comes from employing sand charges only in practice. To plant non-controllable mines one vessel is necessary; mechanical arrangements are made on board so that the mines with their anchors may be dropped at desired intervals as the planter runs a course; the proceeding is extremely dangerous. To plant electrically-controlled mines three vessels at least are necessary—a mine planter, a distribution-box boat and a small power launch as a marking boat. A tug one hundred and fifty feet in length and twenty-five to thirty feet beam will answer as a mine planter. Large deck space forward and little rigging to the rails are desirable. Fig. 4 shows a planter with a mine prepared for launching. Good steam or electrical winches are necessary.

The distribution-box boat should be short and broad of beam. A raft made by flooring over two sand scows may be used for this purpose in emergency. A powerful gasolene launch, about thirty feet long should be used as a marking boat.

The mines, anchors, mooring rope and other appliances are piled conveniently on the forward deck; the mine cables attached to the mines carried aft and coiled in the form of figures of eight piled on top of each other in the proper order. A group of mines is shown in Fig. 5.

The center mine is planted first and the others alternately from the port and starboard sides. A buoy is dropped to show where the first one is to go, the power boat ties to the first mine planted and acts as a marker for the next two; then it picks up the buoys of the mines successively so as to act as a marker throughout the entire operation. To plant a mine the mine planter passes near the distribution-box boat, throws

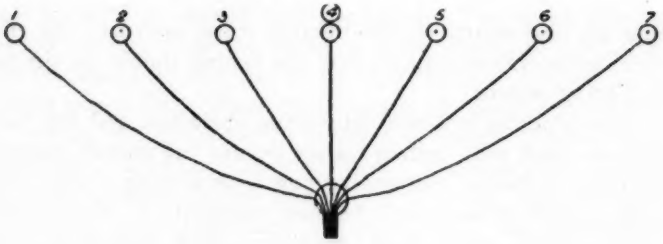


Fig. 5.

a cable to it, and proceeds to the point where the mine is to be planted. The cables pay out, and when the desired spot is reached the mine and anchor are both dropped at the command "let go;" best results are obtained by dropping the anchor a trifle sooner than the mine.

A channel may be closed in this manner at the rate of eighteen hundred feet in two hours and a half; therefore one planter with its small boats can close one and a third miles of channel per day. This is under the supposition that everything is ready for operation when the order is given.

The mine field is but one of the elements of harbor defense, and these are all so related and interdependent that it is necessary to consider each to a greater or less degree to appreciate the true part that any one of them plays.

In the evolution of coast-defense appliances the direct-fire gun was naturally the first to be developed, and all energies were spent toward that end. As time progressed certain other elements appeared, but until recently have been considered and classified as auxiliary. One of the fundamental laws of evolution is that it proceeds with rapid strides when conditions are ripe; this applies with wonderful exactness to the so-called auxiliary during the last decade. In fact, so rapid has been the progress that today the main elements of harbor defense are direct-fire guns, mines (including torpedoes), mortars and searchlights; the first being given precedence only by virtue of its age. If we consider their moral effect, their present state of efficiency and accuracy and the damage when success-

ful, the mine and mortar may be considered the most important factors in deterring an enemy from attacking a fortified harbor. However, it is considered that the functions of all these elements are so coördinate that each must be rated of prime importance, and incomplete without the coöperation of the others. To appreciate this we have but to consider the uselessness of guns—including mortars—at night without their eyes in the form of searchlights, and, in turn, of the uselessness of searchlights in smoke, fog and mists, in which case the defense would be practically helpless without mines; these, on the other hand, must be protected by guns during the period preceding the mist or fog under cover of which an attacking fleet would attempt to force a passage. Period preceding is used advisedly, for modern mine fields cannot be tampered with sufficiently during the fogs and mists to make a passage through which a fleet of battleships would attempt to pass. It should not be necessary to emphasize the absolute necessity of mines, to anyone who has observed the ease with which ships navigate the channels of our northern harbors through a mist so thick that range finders cannot be used.

The interdependence of guns and mines was recognized, though not acted upon, years ago. In the report of the Endicott Board the following principles were enunciated: "Without powerful guns in the defense the armored ships of the enemy would proceed deliberately to the removal of the mines; and without the aid of mines the enemy's vessels could not be prevented, generally, from running past the batteries."

This statement is true provided the defense is supplied with the means of meeting an attack of mine fields made by modern submarine boats, and it may be asserted confidently that, so long as the gun defenses are efficient, the mine fields cannot be disabled seriously, and so long as the mine fields remain in operation no naval commander will ever risk voluntarily a battleship, the mainstay of his country's naval strength, and an element whose loss would be irreparable within the duration of a modern war, in a hazardous attempt to pass such a field.

In a paper on ships and fortifictions a prominent naval authority says, in effect, that fleets will not attack alone harbors fortified properly, and that their use in such attacks will be simply supplementary to attacks from the land side. Fortified properly, in the statement, had special reference to mines and mortars.

Let us now consider the mine defense. The elements of this are fixed mines, previously described, and automobile torpedoes. Mines became practical and valuable, though very uncertain, factors of defense, in the Civil War. The limited success obtained with them, however, did not seem to produce much effect, as no nation since that time has shown in warfare that she has a submarine defense worthy the name, and some have shown hopeless grasp of the possibilities of torpedoes and mines. The wonderful efficiency of these weapons in destroying vessels has been well demonstrated in the recent war in the East. There mechanical mines were employed. These, owing to their cheapness and quickness of planting, appeal to some, but their uncertainty, their danger to friend as well as foe, and the lack of knowledge of their state and position after their planting, make their use prohibitive except in harbors of no importance and no commerce.

The splendid results of a comparatively inefficient mine service in the hands of the Confederates was a fine example of what may accrue to a weak adversary from the efficient use of a mine system. Furthermore, an effective gun and mine defense of a harbor is a source of great strength to a navy, for the prime objective of the latter is the enemy's fleet. A fleet is essentially an offensive weapon, and a country whose policy calls for detachments from its fleets to protect its ports can be condemned only in the strongest terms. Naval offense is the paramount consideration in any successful war, naval defense the most expensive and precarious undertaking, and has only one conclusion well exemplified in the last two wars. Anything which in the smallest way contributes to the former and relieves the latter may be considered a most valuable asset to any country. With our shore defenses per-

fected our navy may cut loose and carry war wherever strategic reasons may require.

The only safe and effective mine system is the electrically-controlled. With such a system the mine fields may be kept in repair, the operator has absolute information as to the condition of every mine and the mine may be made perfectly safe for friendly vessels; these characteristics are the prime considerations.

The cost of mine defense is a mere bagatelle when compared with that of any other. The entire material for all harbors of a country like ours will cost little more than half that of a single battleship, viz: about $4,000,000.

The personnel necessary for keeping the material in perfect condition, planting at a moment's notice and operating, is about five thousand officers and men. I have stated here, planting at a moment's notice under modern conditions; this is really the determining factor of an efficient mine defense.

The service of the mine fields includes not only planting, repairing and operating the mines, but also serving rapid-fire guns for mine field protection. The mine organization for a typical harbor contemplates a mine commander who will have command of all personnel connected with mine operation and the protection of the mine field. He will have a station in electrical communication with his rapid-fire guns, his searchlight and his operating room, so that he may exercise control readily over all of the units of his command. He would require as his assistants two officers for the service of the mine field, two for rapid-fire guns and two for operating the system. After the mines are down, enough men will be liberated from the planting detachment to form an effective nucleus for manning the rapid-fire guns to protect the field. A careful consideration of all the difficulties to be encountered by small boats in attempting an attack on a mine field will show that the number of rapid-fire guns for its protection will bear only a small ratio to those that will be necessary for action against small craft which may attempt to force a passage into a harbor.

Mines may be planted with facility in water whose depth is two hundred feet, and may be planted in deeper water, but the difficulties increase very rapidly with the depth. Damages to a mine field from attack or other cause may be repaired readily by planting in rear of the opening due to the disabled mines.

It is at present impracticable to defend by submarine mines certain of our harbors and waterways (few, it is true, yet of the utmost importance) owing to excessive width or great depth of the channel, or to unusual swiftness of the current (these factors existing singly or in various combinations). In such situations the automobile torpedo is a necessary adjunct to the mine field. There are certain limitations to the use of this weapon.

1st. Its range is restricted; therefore it is unsuited to a channel over two and one-half miles in width.

2d. It is launched from a tube either beneath the surface of the water or at a height not greater than nine feet above the surface, and for success the water beneath the point of launching should be at least thirty-six feet deep. This last requirement restricts the use of such torpedoes to those situations where the water near the shore is of unusual depth, or else involves a construction of costly and bulky launching platforms, ordinarily requiring expensive artificial protection. Therefore, these torpedoes cannot be employed, in general, where the channel is very wide or where the shores are low and where there is an extent of shallow water between the shore line and the requisite deep water. Furthermore, their accuracy decreases rapidly with increased range and their range is quite limited, hence for wide channels the advisability of using them from shore stations is much to be doubted.

Full consideration of the various difficulties involved in operating effectively the launching of automobile torpedoes led the Board of Engineers, in August, 1902, to suggest the use of scows for firing platforms, rejecting all ideas of shore stations proper.

Reflection will show that in order to protect such scows

they must be converted into armored vessels, and that with the exception of slight mobility nothing would be gained. Therefore we conclude that the proper launching platforms for the automobile-torpedo defense of a harbor are the torpedo boat and submarine boat; the latter is the ideal coast-defense launching platform for this purpose. It possesses in a high degree the valuable properties of mobility, concealment and efficiency, and in addition it may, within limits, select the most advantageous time and point of attack.

Besides this particular use the submarine boat will prove undoubtedly of great value in harbor defense for the following purposes:

1st. As a means of keeping a blockading fleet at a great distance from the harbor entrance at night and also preventing hostile ships from anchoring at any time in the vicinity of the entrance.

2d. As outposts and scouts to prevent interference with the mine field by surface boats in foggy weather or by submarine boats at any time.

For attaining these objects only the smaller or harbor type of submarine is necessary. The uses outlined here for submarine boats as adjuncts to fixed coast defenses are purely defensive or defensive-offensive, and are distinct from the essentially offensive use to which such boats will be put by an enterprising navy.

The zone of activity of the fixed defenses of a harbor is that covered by their heavy-gun fire, and within this zone there can be no division of authority; that is, the commander upon whom the responsibility of the defense devolves must have unquestioned and absolute control of all the defensive elements. Therefore it follows that where such boats are provided they should be made an integral part of the artillery district command, not separated therefrom except in great emergency, and then only by the highest authority.

The present type of submarine is purely a harbor-defense boat and may be called a local issue.

It is not contended here that the army should furnish the

officers and supply of these boats, but that they are part and parcel of the proper defense of an artillery district, and should be under the district commander for tactical use. They are on the border line between the coast artillery and naval services. The navy graduates into them in its offensive-defensive rôle, the coast artillery does likewise in its defensive-offensive operations.

Having outlined the main elements of the defense and given somewhat in detail the factors of the submarine part, we will consider their relation in battle. The battle area for heavy guns extends to 12,000 yards. If mortars are located on the main line of defense they can cover any of the area from the outer limits to a point two thousand yards from the main line. In the outer limits little is to be feared from the ship, but, all things considered, the shore guns have some chance of disabling the enemy; within the destructive area from four to eight thousand yards, the rapid-fire guns of the fleet will not be effective, but the heavy guns of the defense will be destructive if concentrated on the enemy; the rapid-fire guns of the defense will be effective likewise against small vessels in the nearer portion of this area. Within four thousand yards the rapid-fire guns of the enemy will be effective, as well as the high-power guns.

The mine field must be located tactically with the foregoing in view; the other factors involved in the problem, being local, to wit: width of channel, depth of harbor, swiftness of current.

We find nearly all writers asserting that mine fields may be put out of action—

1st. By grappling for cables and junction boxes.

2d. By countermining.

3d. By sweeping.

I wonder how many authorities who have made these statements ever tried any of these operations. I will venture to say none. I have tried some of them under the most favorable circumstances, viz: in broad daylight with a calm sea; even under these conditions very little was accomplished in a

long time. I have also tried them at night when I knew just where the mines were, and it was a hopeless task. Such has been my experience that I am willing to assert positively that nothing is to be feared in fair weather from an enemy who tries to disable a mine field by any of these methods, so long as there is any shore defense at all. In case of moderately thick weather something might be accomplished, but I am convinced that the mine field *in toto* cannot be put out of action so long as the shore defense offers any resistance.—"Journal of the U. S. Artillery."

SPEED TRIALS AND SERVICE PERFORMANCE OF THE CUNARD TURBINE STEAMER *LUSITANIA*.*

By Thomas Bell.

The following hourly abstract on one of the watches on the *Lusitania* brings home to one's mind the loss in steam and speed caused by cleaning fires, especially when the coal is small. It can be easily calculated from this what an appreciable increase in the ship's mean speed could be obtained from this cause alone, if the price and supply would admit of the use of some system of oil-fuel burning:

.....	Mean Revolutions.	Corresponding speed of ship.	
First hour	178	about 24.0 knots	24.15
Second hour	181	about 24.3 knots	
Third hour	186	about 25.0 knots	25.05
Fourth hour	187	about 25.1 knots	
Mean for watch	183	about 24.6 knots.	

Regarding the observations from readings taken in the engine room on the official trials generally, it may be stated that on the measured miles the revolutions were obtained from electric records in connection with the pallograph apparatus, but on the lengthened trials they were taken from half-hourly readings of the engine-room counters. The vacuum recorded is that of the vapor in the main exhaust orifice forming the top of the condensers, and as measured by a siphon mercury gauge, the readings of which throughout are corrected to correspond to a 30-inch barometer. The total quantity of feed water is obtained from hourly counter read-

* Read before Institution of Naval Architects, April 9, 1908.

ings of the double strokes of the Weir's feed pumps, the average length of stroke and the slip or leakage of each pump being determined, both before and after the trials, by careful tests.

The consumption of steam of the auxiliary machinery is obtained by noting the amount by which the temperature of the total feed water was raised in the feed heaters, and to the amount thus found must be added the steam used in the turbo-generators, the exhaust from which was led direct to the auxiliary condensers on the official trials. As before stated, in actual service these turbo-generators exhaust into the contact heaters, and thus raise the feed temperature to about 200 degrees. These connections had to be slightly altered at the time of the trials and, unfortunately, therefore, advantage could not then be taken of this additional source of economy.

The torque horsepower was obtained by the Denny-Johnson apparatus, and the records show that, while a propulsive efficiency of the whole installation was obtained which accorded with the original estimate, the steam consumption of the turbines themselves was very satisfactory. It need hardly be pointed out that those two, viz: propulsive efficiency and steam consumption per unit of power, form an excellent check on each other, for whatever would unduly favor one would be at the expense of the other.

Table I.

Time.—Double runs.	Pressures.		Vacuum at 30-in. barometer.	Revs. per minute.	Speed, in knots.	Shaft horsepower.	Slip of propellers, per cent.
	H.P. receiver, pounds.	L.P. receiver, pounds.					
First	157	5½ pounds	28.0″	194.3	25.62	76,000	17.2
Second	135	2½ pounds	27.9″	186.0	25.0	65,500	15.5
Third	110	½ pound	28.1″	174.2	23.7	51,300	14.5
Fourth	90	3½″ vac.	28.1″	161.5	22.02	40,500	14.3
Fifth	70	6½″ vac.	28.0″	147.6	20.4	29,500	13.1
Sixth	50	10½″ vac.	28.0″	131.1	18.0	20,500	13.7
Seventh	35	14½″ vac.	28.1″	116.1	15.77	13,400	14.6

The *Lusitania* was floated out of dry dock at Liverpool on July 22, 1907, and was thereafter coaled by the Cunard Company, the bunkers for the forward and after boiler rooms being filled with South Wales coal, and those of the two middle boiler rooms with Yorkshire coal.

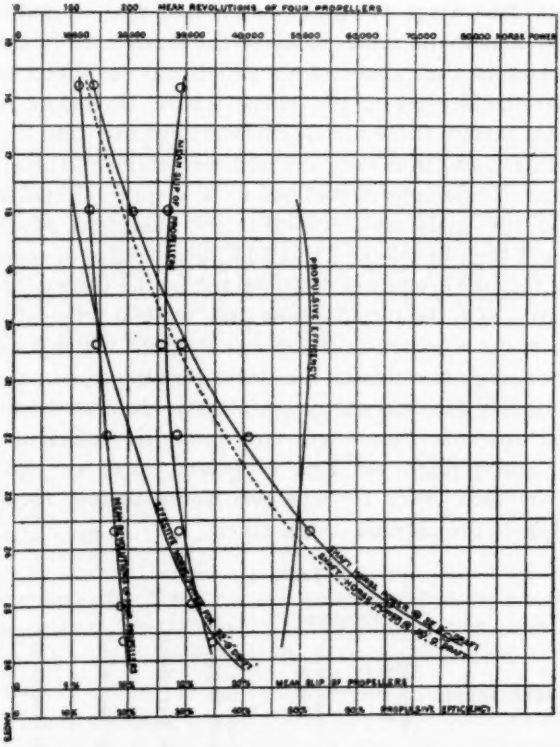


Fig. 5.—PROGRESSIVE TRIAL, JULY 27, 1907, OFF SKELMORLIE.

On her return to the Clyde, on the morning of July 27, a series of progressive runs was made on the Skelmorlie measured mile, as recorded on Table I, with the ship at a mean draught of 32 feet 9 inches, corresponding with a displacement of 37,080 tons. These results are also given in graphical form in the diagram on Fig. 5, which gives curves of

shaft horsepower, revolutions and slip on a common speed base. Two other most interesting curves have been added, one showing the effective horsepower determined by means of tank experiments, and the other showing the propulsive efficiency. It was intended to repeat this trial at the termination of the official trials, at a mean draught of about 30 feet. Unfortunately, however, thick weather on the morning of August 2 prevented this being carried out, but the dotted curve on Fig. 5 indicates with sufficient accuracy what might have been expected.

On the evening of the 27th the *Lusitania* proceeded on a pleasure cruise around Ireland, during which consumption trials at 18, 21 and 23 knots were carried out, and, after landing the guests in the forenoon of the 29th, the vessel returned to the Clyde, making a consumption trial at 15¾ knots *en route*, the results of these trials being given in the first four columns of Table V. After checking the draught of ship, etc., the 48-hours' full-speed continuous trial was commenced at midnight. This trial consisted of two double runs on a course of 304 nautical miles between Corsewall Point and the Longships, and the results obtained are recorded on Table II and the last column of Table V. The mean draught at starting was 32 feet 7 inches, and at the finish about 30 feet 8 inches, the mean for the run having been 31 feet 7½ inches, corresponding with a displacement of 35,600 tons.

Table II.

Time.	Pressures.		Vacuum at 30-in. barometer.	Revs. per minute.	Speed, in knots.	Shaft horsepower.	Slip of propellers, per cent.
	H.P. receiver, pounds.	L.P. receiver, pounds.					
First run	146	2½	27.9″	188.8	26.35	70,400	...
Second run	145	2½	28.0″	187.4	24.3	68,200	...
Third run	146	2½	27.9″	187.5	26.3	68,700	...
Fourth run	148	2½	27.8″	187.9	24.6	68,100	...
Mean of means	146	2½	27.9″	187.9	25.4	68,850	15

The coal consumed in the fifty hours during which the engines were running at full speed was found by measurement of bunkers to be about 2,200 tons. This represents an evaporation of 10.1 pounds of water per pound of coal from 165 degrees temperature of feed, or 11.1 pounds from and at 212 degrees, and a consumption of coal for all purposes of 1.43 pounds per shaft horsepower per hour, with a rate of combustion of coal of 24.3 pounds per square foot of grate surface per hour. The number of stokers on watch was the same as in actual Atlantic service, and the air pressure in the ashpits did not exceed $\frac{3}{4}$ inch of water column. The port evaporators were used for ten hours of the trial; but, as the vapor from these was condensed in the port auxiliary condenser, to which the exhaust from one set of the turbo-generators was led, they were discontinued and the make-up feed obtained from the reserve tanks for the remainder of the time.

The third trial, recorded on Table III, which was commenced in the forenoon of August 1, consisted of one full-power double run between Corsewall Point and the Chicken Rock, a distance of 59 nautical miles each way, but comparison with the dotted curve on Fig. 5 shows that the tide conditions during this trial give altogether too favorable a speed

Table III.

Time.	Pressures.		Vacuum at 30-in. barometer.	Revolutions per minute.	Speed, in knots.	Shaft horsepower.	Slip of propellers, per cent.
	H.P. receiver, pounds.	L.P. receiver, pounds.					
First run.....	152	2¼	28″	191.3	26.75	72,000	.....
Second run.....	152	2¼	28″	191.7	26.17	72,800	.....
Mean.....	152	2¼	28″	191.5	26.46	72,400	13.2

result. The mean draught was 30 feet 4½ inches, corresponding with a displacement of 34,160 tons. The vessel was then headed for Ailsa Craig, and carried out the specified six full-power runs between Ailsa Craig and the Holy Isle (off Arran).

These latter, recorded on Table IV, which were run at a mean draught of about 30 feet 2 inches, corresponding with a displacement of 33,770 tons, give a very reliable record of power and speed at this draught, when compared with the 25.62 knots obtained on the measured mile at 32-feet 9-inch draught. On the following day weather conditions precluded any further trials, and the reversing trial and the steering and circle-turning trials were accordingly carried out on the vessel's passage to Liverpool on August 26.

In a fast passenger liner, such as the *Lusitania*, it is of the utmost importance that the maneuvering capabilities should leave nothing to be desired, and to demonstrate the possibilities of the ship in this respect, various trials were made, the most important being the following:

Stopping trial.—The ship was run on the Skelmorlie measured mile at a speed of 22.8 knots, the average revolutions of the propellers being 166 per minute. On entering the mile, the engine-room telegraphs were rung to "full speed astern;" the ship was brought to rest in 3 minutes 55 seconds, the distance run being about three-quarters of a mile, or about six times the length of the ship. During this trial the boilers in the three after boiler rooms only were in use, and the initial pressure at the astern turbine was about 90 pounds per square inch.

Table IV.

Time.	Pressures.		Vacuum at 30-in. in barometer.	Revolutions per minute.	Speed, in knots.	Shaft, horsepower.	Slip of propellers, per cent.
	H. P. receiver, pounds.	L. P. receiver, pounds.					
First run	151	2¾	28″	191.3	25.62	72,500	...
Second run	152	2¾	28″	191.1	26.36	72,000	...
Third run	153	3	28″	191.9	25.31	72,000	...
Fourth run	147	2½	28″	191.0	26.16	72,100	...
Fifth run	149	2½	28″	190.2	25.26	70,800	...
Sixth run	149	2½	28″	191.2	25.95	71,800	...
Mean of means	150	2¾	28″	191.2	25.77	71,910	15.3

Circle trials.—With the ship initially on a straight course, and the turbines running at an average speed of 180 revolutions per minute, the steering wheel was put hard over in 17 seconds. The tiller went over to 35 degrees in 20 seconds, and the vessel made a complete circle in 5 minutes 50 seconds, the average revolutions coming down at the completion of the circle to 70 per cent. of the rate at the commencement. The resulting circular path was approximately 1,000 yards, or four lengths of the ship, in diameter. This maneuver was made both under starboard and port helm with very closely confirmatory results.

Going astern with the inner propellers running at a uniform rate of 136 revolutions per minute and under full helm, resulted in half circles being made in an average time of 6 minutes 45 seconds.

Table V.

ACTUAL STEAM AND COAL CONSUMPTION OF MAIN AND AUXILIARY ENGINES AT VARIOUS SPEEDS UNDER CONDITIONS PREVAILING ON OFFICIAL TRIALS, VIZ: TURBO-GENERATORS EXHAUSTING TO AUXILIARY CONDENSERS, OTHER AUXILIARIES EXHAUSTING TO HEATERS.

Shaft horsepower	13,400	20,500	33,000	48,000	68,850
Speed, in knots	15.77	18.0	21.0	23.0	25.4
Temperature of feed water	200°	200°	199°	179°	165°
Total consumption of auxiliaries, in pounds per hour	71,000	76,400	85,700	96,700	116,500
Total consumption of turbines, in pounds per hour	284,500	353,600	493,300	668,300	879,500
Steam consumption of auxiliaries, in pounds per turbine horsepower hour	5.3	3.72	2.6	2.01	1.69
Steam consumption of turbines, in pounds per horsepower hour	21.23	17.24	14.91	13.92	12.77
Total steam consumption, in pounds per horsepower hour	26.53	20.96	17.51	15.93	14.46
Coal consumption, in pounds per horsepower hour	2.52	2.01	1.68	1.56	1.43
Estimated coal consumption, in tons, on a voyage of 3,100 nautical miles, allowing 20 tons for galleys, etc	2,980	3,190	3,670	4,520	5,390

As important factors contributing to these very satisfactory results, it may be remarked that, following on the suggestion of Sir Philip Watts, the deadwood aft is cut away in a fashion

similar to that in recent warships. The inner propellers are fairly close together and, as the rudder is of large dimensions in the fore-and-aft direction, the race from these propellers impinges fully upon it when any helm is used. The vessel, consequently, is very similarly circumstanced to a single-screw ship, or a triple-screw ship with all three propellers in action, and gets steerage way without any perceptible headway, and this feature was very noticeable during the steering trials. At first sight it would appear that the outer propellers should have been those utilized for maneuvering purposes, as the outer shafts have about three times the spread from the middle line that the inner shafts possess. For turning with propellers alone without the help of the rudder they would have been much the more effective, but they would not have possessed any such advantage as that alluded to above in respect to obtaining steerage way without headway.

Table VI.

ESTIMATED STEAM AND COAL CONSPUMPTION AT VARIOUS SPEEDS, ALLOWING FOR THE ADDITIONAL AUXILIARY STEAM CONSUMPTION FOUND REQUISITE UNDER ACTUAL SERVICE CONDITIONS FOR THE WASHING-WATER SUPPLY, ETC., WITH A FULL COMPLEMENT OF PASSENGERS, WEATHER CONDITIONS BEING AS ON OFFICIAL TRIAL.

Shaft horsepower	13,400	20,500	33,000	48,000	68,850
Speed, in knots	15.77	18.0	21.0	23.0	25.4
Temperature of feed water	200°	200°	200°	200°	200°
Total consumption of auxiliaries, in pounds per hour	93,500	100,900	112,700	127,500	149,700
Total consumption of turbines, in pounds per hour	284,500	353,600	493,300	668,300	879,500
Steam consumption of auxiliaries, in pounds per turbine horsepower hour	6.97	4.92	3.41	2.65	2.17
Steam consumption of turbines, in pounds per horsepower hour	21.23	17.24	14.91	13.92	12.77
Total steam consumption, in pounds per horsepower hour	28.2	22.16	18.32	16.57	14.94
Coal consumption, in pounds per horsepower hour	2.76	2.17	1.8	1.62	1.46
Estimated coal consumption, in tons, on a voyage of 3,100 nautical miles, allowing 20 tons for galleys, etc.	3,270	3,440	3,930	4,700	5,490

Table VI has been compiled for comparison with Table V, to show the additional consumption of steam for auxiliary purposes under actual working conditions at sea with the ship full of passengers. This shows very clearly the demand which modern improvements make on the steam, and hence coal consumption of a large passenger vessel. An additional line has been added to Tables V and VI to show total coal consumption on a voyage of 3,100 nautical miles at the various speeds and under the different conditions.

Table VII.

ABSTRACT OF ENGINE-ROOM LOG FOR THIRD VOYAGE WEST: QUEENSTOWN TO NEW YORK.

Date when last dry docked, July 22, 1907. Mean draught, leaving Queenstown, 33 feet 7 inches. Mean draught arriving New York, 30 feet 10 inches.

Date, 1907.	Steam Pressures.			Temperatures.		Vacuum, inches.	Barometer, inches.	Length of day.		Distance by observation, nautical miles.	Mean speed.	Mean revolutions.	Mean slip.	Coal consumed for main and auxiliary engines.
	Boilers, pounds.	H.P. recrs., pounds.	L.P. recrs., pounds.	Hotwell.	Feed water.			H.	M.					
Noon,													*P.ct.*	*Tons.*
Nov. 3	170.0	140.0	2.3	68°	200°	28.0	30.4	...	52	21	24.24	182 5	16.5	40
Nov. 4.....	169.1	142.2	2.2	78°	197°	28.0	29.7	24	57	606	24.28	182.6	16.4	1,090
Nov. 5.....	167.3	140 6	2.3	78°	198°	28.2	30.0	25	2	616	24.6	182.8	15.4	1,090
Nov. 6.....	168.3	140.4	2.5	70°	196°	28.2	30.1	24	55	618	24.8	183 5	15.1	1,090
Nov. 7.....	168.3	138.3	2.2	72°	195°	28.0	29.6	24	52	610	24.52	181.4	15 0	1,090
1.14 a. m.,														
Nov. 8.....	165.0	132.5	1.5	75°	200°	27.8	29.3	14	2	310	22.09	174.0	20.2	*576
Means.....	168.0	139.3	2.2	74.5°	197°	28.1	29.8	...	...	...	24.25	181.1	15.9	4,976
Total.....	...	...	...	...	...	...	...	114	40	2,781	...	...	...	...

* This includes all coal used till 10 a. m. on the 8th.

Summary of total coal consumed on voyage: Liverpool to Queenstown, 408 tons; Queenstown to New York, 4,976 tons; galleys, etc., 18 tons; total coal taken from bunkers, from leaving landing stage, Liverpool, till moored at wharf, New York, 5,402 tons. Passage—Queenstown to Sandy Hook—4 days, 18 hours, 40 minutes.

With reference to the third voyage west, from November 2 to November 8 of last year, thanks to the courteous permission of the chairman of the Cunard Company, the leading particulars of the official engine-room log are summarized in Table VII. Regarding the mean draught of the vessel at sea, it may be remarked that, after the second day out certain of the forward tanks were gradually filled for the purpose of avoiding excessive trim, so that the mean draught on Novem-

ber 5, 6 and 7 was approximately 32 feet, or very little more than the mean of the first pair of runs from Corsewall Point to the Longships and back. The conditions, however, were otherwise very different, for, with the exception of the twelve hours of fine weather and smooth sea from noon till shortly after midnight on November 6, it was throughout the average mid-Atlantic winter weather—namely, strong winds and resulting boisterous sea. Up till midnight on the 6th, *i. e.*, for 2,176 out of a total of 2,781 nautical miles, the mean speed works out at 24.65 knots; but, unfortunately, early on the 7th the wind freshened, gradually increasing to a furious southwest gale, which reached its height about 4 P. M., and reduced the average speed for the last 24 hours below 23 knots, and thus brought down the mean average for the com-

Table VIII.

Date—1907.	Length of steaming day.		Distance run, nautical miles.	Speed, knots.	Total distance steamed.	Total time.		Mean average speed.
	Hrs.	*Min.*				*Hrs.*	*Min.*	
Noon, November 3.....	0	52	21	24.24	21	0	52	24.24
Noon, November 4.....	24	57	606	24.28	627	25	49	24.27
Noon, November 5.....	25	2	616	24.6	1,243	50	51	24.44
Noon, November 6.....	24	55	618	24.8	1,861	75	46	24.57
Noon till midnight, November 6.....	12	30	315	25.2	2,176	88	16	24.65
Noon, November 7.....	12	22	295	23.85	2,471	100	38	24.55
Morning, November 8.	14	2	310	22.09	2,781	114	40	24.25

pleted voyage to 24.25 knots. Table VIII, giving the mean average speeds at the different stages of the voyage, shows very clearly the effect of this gale, unfortunate so far as preventing the vessel from complying with the contract conditions, but giving those connected with the ship an opportunity of thoroughly satisfying themselves as to her behavior when driving through the huge waves at about 22½ knots, without any racing of engine or sign of laboring, and dis-

pelling the idea, current in some minds, that turbine-propelled ships do not show to advantage in heavy weather.

The following particulars of the steam consumption are given in conjunction with the figures of coal consumption set forth in Table VII. Throughout the voyage a careful record of the feed-pump counters gave an average of 998,000 pounds of water pumped into the boilers per hour. Of this, about 114,000 pounds was used by auxiliary machinery exhausting into the feed heaters, 26,000 pounds by the evaporating plant supplying feed make-up and washing water, and about 6,500 pounds for steam to the thermotanks, galleys and pantries, both of which latter figures are based on data obtained from tests carried out before the vessel left the Clyde. Hence, taking the average shaft horsepower as 65,000, the steam consumption per shaft horsepower per hour works out as follows:

	Total water.	Per shaft H.P. hour.
Main turbines	851,500 pounds =	13.1 pounds.
Auxiliary machinery	114,000 pounds =	1.75 pounds.
Evaporating plant and heating	32,500 pounds =	.5 pound.
	998,000 pounds	15.35 pounds.

Average amount of coal burned per hour for all purposes = 43½ tons.
Water evaporated per pound of coal = 10.2 from a feed temperature of 196°.
Water evaporated per pound of coal = 10.9 from and at 212°.
Coal for all purposes per shaft horsepower, per hour = 1.5 pounds.
Coal per square foot of grate, per hour = 24.1 pounds.

Taking a mean displacement of 36,000 tons, this represents at 24½ knots a consumption of almost exactly 11 pounds of coal per 100 nautical miles per ton of displacement. The coal used was half South Wales and half Yorkshire, practically the same as on the official trials.—"International Marine Engineering."

A METHOD OF PROTECTING FROM CORROSION THE ENCASED PORTION OF PROPELLER AND STERN-TUBE SHAFTS.

By Leo Morgan, Member.

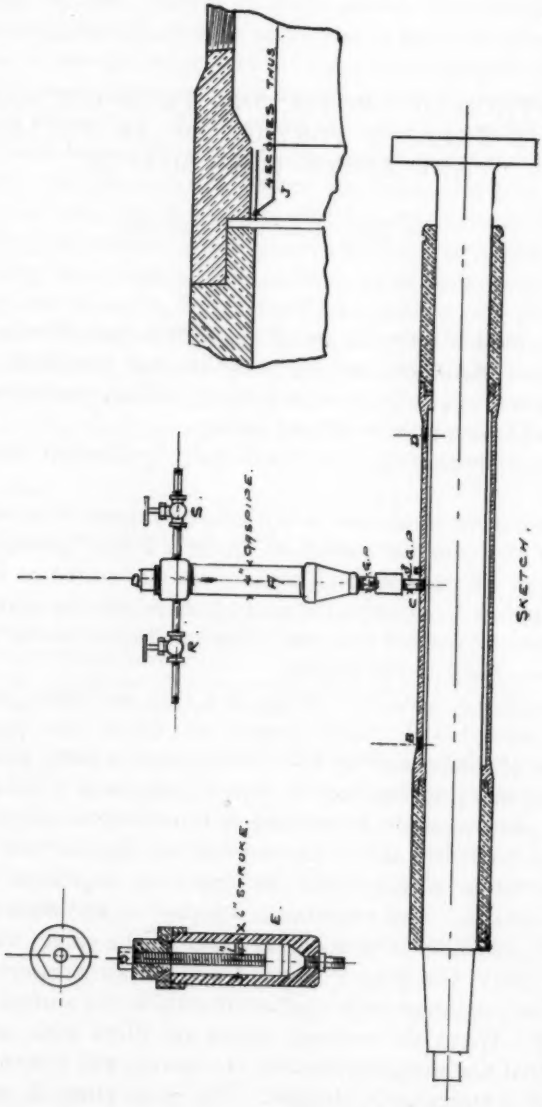
The method now in use at the Union Iron Works, San Francisco, Cal., for making propeller and stern-tube shaft casing watertight consists in forcing red-lead putty into the recesses between the shaft and casing.

The accompanying sketch will help to illustrate this process.

Each recessed section is drilled and tapped at about the center and near the ends B, C, D, for a ⅜-inch gas-tap plug. The end bearings of these sections are also scored at four or more points (J) to allow the putty to flow into the joints. It will also be noticed that the joints stand apart on the inside and are closed on the outside.

The filling material consists of a thin and thick red-lead putty mixed with boiled linseed oil, forced into place by means of compressed air and a hand-pressure putty pump.

The charging chamber A, which consists of a piece of 4-inch gas pipe about 2 feet long, is fitted with a coned cap at at the lower end and at the top with a plug and two ¼-inch globe valves, R and S, for the supply and regulation of the air pressure. This chamber is attached at the center opening C, and filled almost to the level of the valves with the thin putty, the plug P inserted, and an air pressure of 50 pounds per square inch applied directly to the surface of the putty. When the recessed spaces are filled with the thin material the charging chamber is removed and the end holes B and D temporarily plugged. The putty pump E, operated


SKETCH.

by a wrench, is now inserted at C, and the thick putty is forced into the casing until the space in its vicinity is filled solid. The pump is then removed and the hole plugged. This operation is repeated successively at the end holes, B, D, until the entire casing is filled solid, which is easily determined by lightly tapping the casing with a hammer.

In the event of the casing being porous, the putty is forced into the pores and sometimes through to the surface, thereby producing a lighter casing than can be obtained by either the tinning or the shellac treatment, and when properly done will, I think, give satisfactory results.

U. S. SCOUT CRUISER *BIRMINGHAM.*

DESCRIPTION OF MACHINERY AND OFFICIAL TRIALS.

By Commander Theo. C. Fenton, U. S. N., Retired.

The Scout Cruiser *Birmingham* was built by the Fore River Shipbuilding Company, of Quincy, Mass. The contract for this vessel was signed May 17, 1905, the price being $1,556,000. This price does not include the ordnance and ordnance outfit and certain articles supplied by the Government. The contract time for completion was thirty months. Owing to various delays for which the contractors were not responsible, this time was extended to February 11, 1908. The main engines were required to develop 16,000 indicated horsepower when making 200 revolutions per minute, with a steam pressure at the high-pressure cylinder of 250 pounds.

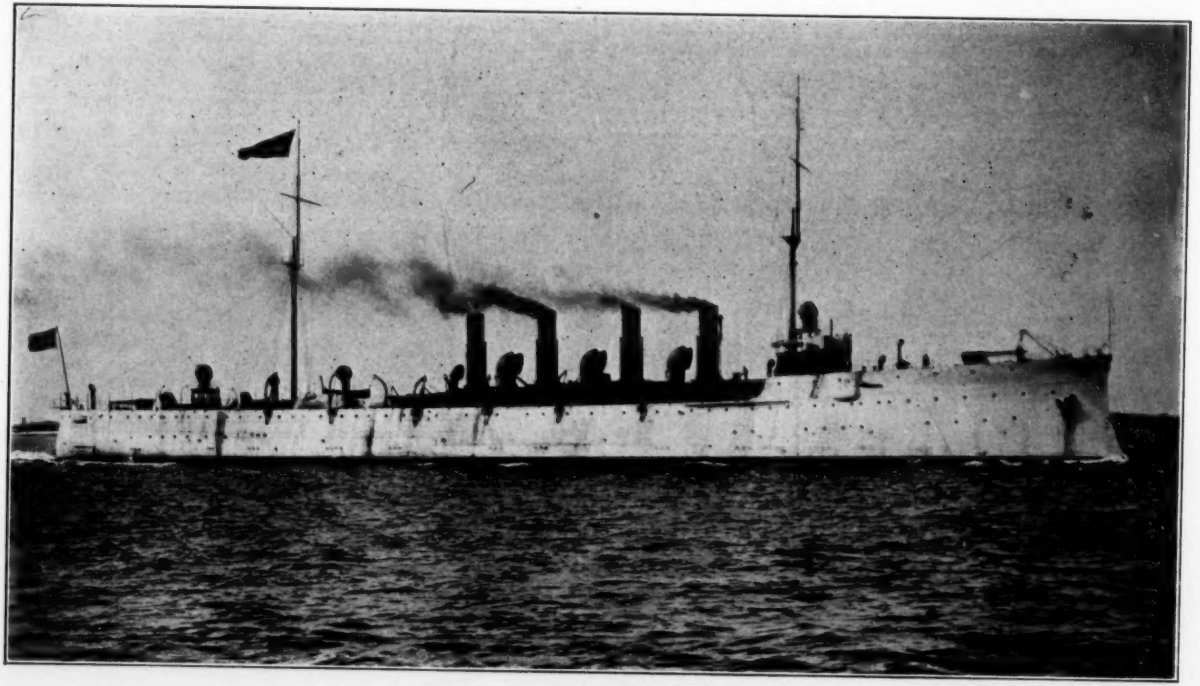
The guaranteed speed of the ship was twenty-four knots per hour for four hours in the open sea.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet	420
over all, feet and inches	423–02
on L.W.L., feet	420
Breadth, molded, feet and inches	46–08
Draught, fully loaded, feet and inches, about	19–01½
Displacement, loaded, tons, about	4,640
Draught on trial, feet and inches	16–09½
Corresponding displacement on trial, tons	3,750
Displacement per inch at L.W.L., tons	31.13
Capacity of coal bunkers (43.5 cubic feet per ton), tons	1,379
engine-room feed tanks, tons	10.65
reserve feed-water compartments, tons	116.46

ELECTRIC PLANT.

There are three generating sets installed in this vessel. They are of 32-kw. capacity, having a full-load output of 256 ampères at 125 volts at a speed of 400 r.p.m., and were



U. S. Scout Cruiser "Birmingham."

manufactured by the General Electric Company, of Schenectady, New York. The generating engines are of the vertical, cross-compound, double-acting, enclosed type, with cranks 180 degrees apart, and are for a normal speed of 400 r.p.m., with a normal steam pressure of 100 pounds, and exhausting into the condenser.

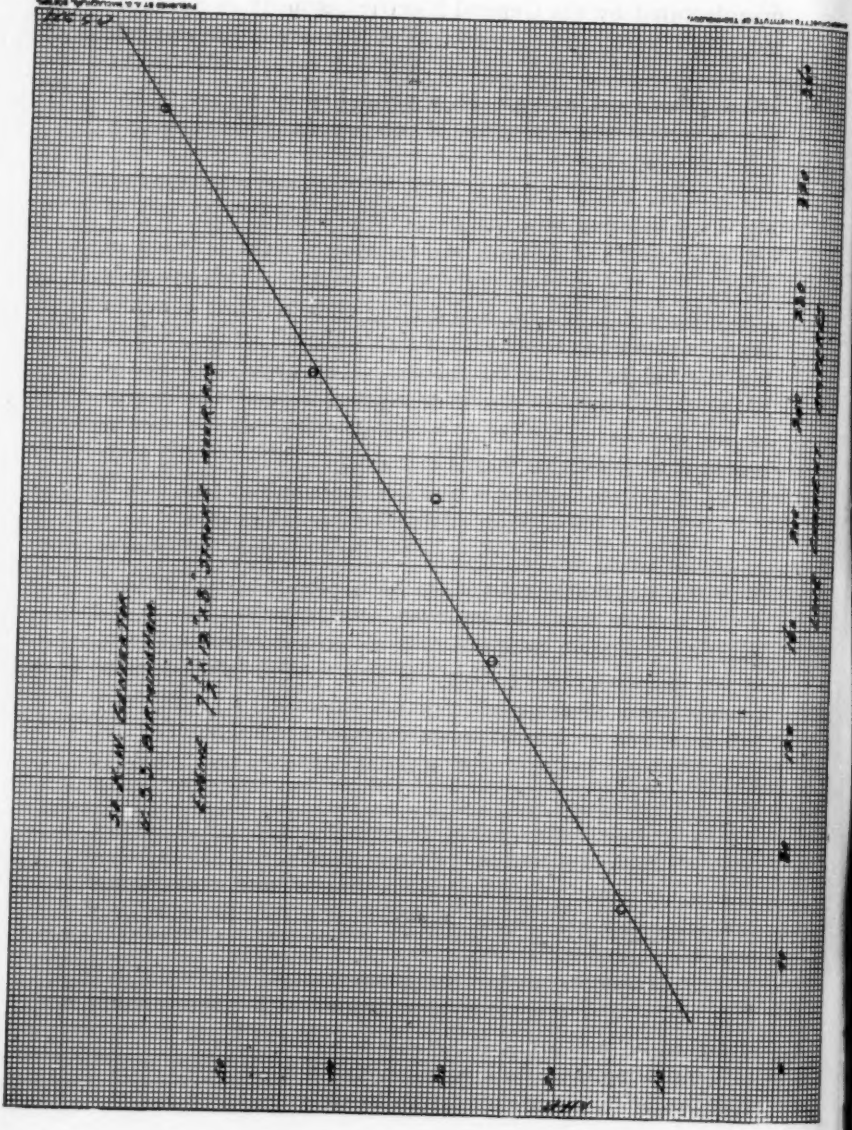
There is a heavy bedplate bolted to the foundation supporting the generating sets, that part of which is under the engine being enclosed and forming the reservoir for the oil used in lubricating the moving parts of the engine. This space is also utilized as a settling and cooling chamber, this chamber being in communication with a sight-tube oil gauge indicating the height of the oil therein. The upper part of the bed-plate casting contains a depression around the engine casing for the collection of waste oil and drippings, and is provided with suitable means of drainage. The moving parts of the engine are accessible through doors on the port side of the casing and the portable plate on the opposite side.

The range of steam pressure for these engines is between 80 and 120 pounds with a normal pressure of 100 pounds. At 80 pounds steam pressure and atmospheric exhaust the generating sets are designed to carry 90 per cent. full load; at 100 pounds steam pressure and atmospheric exhaust, or 80 pounds steam pressure and vacuum exhaust, they will carry full load, and at normal steam pressure and vacuum, or 120 pounds steam pressure and atmospheric, they will carry an over-load of 33 per cent.

The cylinders are $7\frac{1}{2}$ inches and 12 inches in diameter with a stroke of 8 inches. Both cylinders, with their steam chests, are contained in a single casting of hard close-grained cast iron, accurately bored and covered with a thick layer of asbestos lagged with sheet iron.

The generators are of the direct-current, constant-potential, multi-polar type, compound wound with a series shunt, designed for no change in voltage at different loads.

Ventilating fans, deck winches, ammunition hoists and the fresh-water pumps are all driven by electric motors.



ANCHOR WINDLASS.

The anchor windlass is of the Hyde type, manufactured by the Hyde Windlass Company, of Bath, Maine.

The windlass proper is of the vertical type and has two vertical shafts driven by worm gearing direct from a worm located on the crank shaft of the engine, without the intervention of counter shafts or bevel gears. Each shaft carries on its upper end above deck a wildcat with locking gear complete. The arrangement is such that the wildcats can be operated together or independently of each other. The wildcats are revolved in a horizontal plane, taking in each bower chain on the outboard side, and the port sheet chain on the inboard side of the port wildcat.

On the official trials of the vessel this windlass worked in a perfectly satisfactory manner.

STEERING ARRANGEMENT.

The steering gear is located aft in the compartment provided for it, and is of the standard type of the Bureau of Construction and Repair, consisting of a right-and-left-hand screw with traversing ends direct connected by side rods with crosshead on the rudder stock. The weight of the rudder is transmitted to this crosshead by means of a wrought-steel ring fitted in a groove near the head of the rudder stock. Between the crosshead and the casting on the stern post, a floating disk of phosphor-bronze is fitted, recessed in the stuffing-box casting, to hold oil, the weight of the rudder being taken by this floating ring. The casting bolted to the stern post and taking the weight of the rudder through this floating ring is fitted with a stuffing box around the rudder head capable of adjustment from a convenient location in the steering-engine room. A friction band is fitted to the rudder stock and is operated from the steering-engine room.

The steering engine is of sufficient power to put the rudder from hard aport to hard astarboard in twenty seconds when the vessel is moving ahead at full speed, with a working

steam pressure of 150 pounds per square inch. The engine is of sufficient strength, however, to withstand operation under the full boiler power. Provision is made in the steering-engine room for hand steering by wheels on a shaft geared to the main screw shaft. A slip joint is provided between the engine and the screw shaft to take up the lateral motion. Direct hand steering is provided for, in an emergency, by suitable arrangement of relieving tackles direct connected to the crosshead on the rudder stock. The hand-steering wheels are arranged to be thrown out of gear when the engine is at work, and to work readily through the main shaft with the engine out of gear.

PROPELLING MACHINERY.

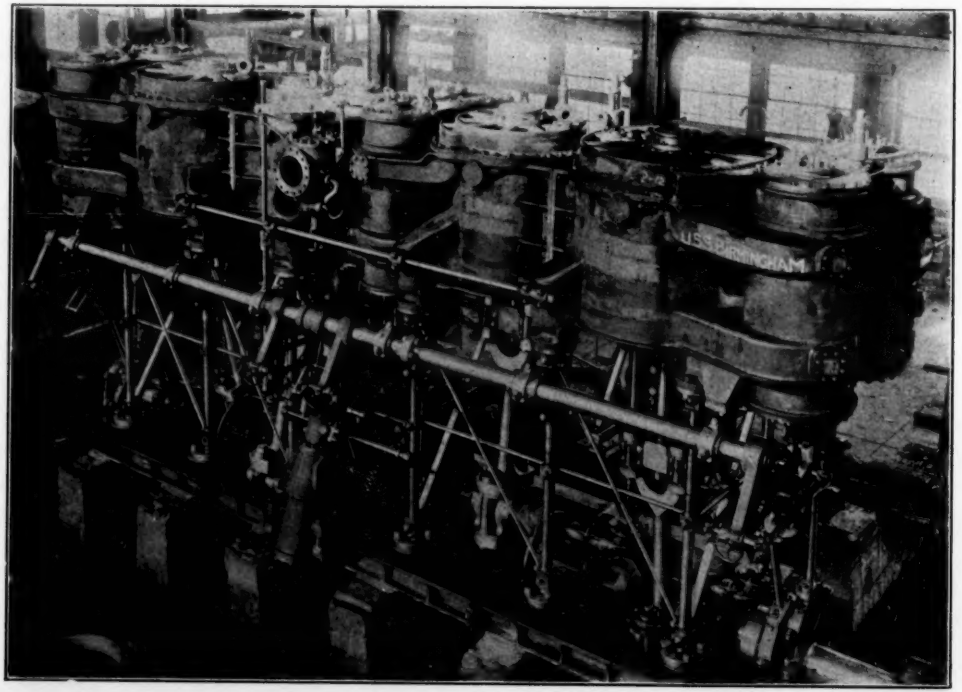
The propelling engines are right-and-left, and are placed in watertight compartments separated by an athwartship bulkhead.

The engines are of the vertical, inverted-cylinder, direct-acting, triple-expansion type.

The order of the cylinders, beginning forward, are forward low pressure, high pressure, intermediate pressure and after low pressure. The forward low-pressure and high-pressure cranks are opposite, also the intermediate and after low-pressure cranks, the second pair being at right angles with the first. The sequence of cranks will then be: high pressure, intermediate pressure, forward low pressure and after low pressure.

The main valves are worked by Stevenson link motions with double-bar links. There is one piston valve for the high-pressure cylinder and two each for the intermediate and low-pressure cylinders.

The frame of the engines consists of forged-steel columns trussed by forged-steel stays. The engine bedplates are of cast steel, supported on the keelson plates. All crank, thrust, line and propeller shafting is hollow. The shafts, piston rods, connecting rods and working parts generally are of forged steel.



ENGINES, U. S. S. "BIRMINGHAM."

REVERSING GEAR.

The reversing gear for each engine consists of a steam cylinder and an oil-controlling cylinder fulcrumed in bearings on the bedplate. The common piston rod of the reversing-gear cylinders acts directly on a forked arm, keyed to the reversing shaft. The piston rod passes through the controlling cylinder with uniform diameter. The valve of the steam cylinder is of the piston pattern, of composition, worked by a continuation of the stem of the steam piston valve. These valves are worked by a floating lever, the primary motion being derived from the hand lever on the working platform, and the secondary motion from the reversing shaft, all parts being so adjusted that the reversing shaft follows the motion of the hand lever and is firmly held when stopped.

A pump for reversing by hand is connected to the oil cylinder, with its lever convenient to the working platform. The by-pass pipes are connected to the valve box of the hand pump in such a way as to leave the hand arrangements always in gear. The piston of the oil cylinder is packed by two cup leathers.

The following are the principal dimensions:

Steam cylinder, diameter, inches	14
Oil cylinder, diameter, inches	8½
Stroke, inches	18

REVERSING SHAFT.

There is one reversing shaft for each engine with an axial hole through it. It has arms for reversing gear and for each link. Each reverse arm for the links is made with a slot fitted with a block, to which the expansion links are attached. Each block is adjustable in the slot of its arm by a screw and handwheel with approved locking device, and is fitted with suitable index. The slot in these arms is so arranged that the links may always be thrown into full backward gear irrespective of the position of the block in the slot; and the length of the slot is such that the cut-off may be varied from about 0.5 to 0.75 of the stroke.

TURNING ENGINES AND GEAR.

There is in each engine room a double engine for turning the main engines with steam of 100 pounds pressure. This engine drives, by worm gearing, a second worm which may be made to mesh with a worm wheel fitted on the crank shaft.

The turning engines have piston valves, and are made reversible by means of a change valve moved by a screw and handwheel. Each engine is fitted for turning by hand.

PROPELLERS.

There are two 3-blade propellers, both outboard turning for ahead motion.

The bolt holes in the flanges of the blades are made oval, to allow of adjustment of the pitch, and each blade is firmly bolted to the hub by tap bolts, secured from turning by lock plates.

The dimensions of the propellers are as follows:

Diameter, feet and inches	12–06
Pitch, as set, mean, feet and inches	15–03
adjustable from, feet and inches	14–06 to 16–06
Ratio of diameter to pitch $\left(\frac{P}{D}\right)$	1.22
Area, projected, square feet	40.8
helicoidal, square feet	49.437
disk, square feet	122.72
Height of lower tip of blade above keel, inches	13
Immersion of upper tip of blade at low draught, inches	46⅛

ENGINE DATA.

Cylinders, number for each engine	4
H.P., diameter, inches	28¼
I.P., diameter, inches	45
F.L.P., diameter, inches	62
A.L.P., diameter, inches	62
stroke of all pistons, inches	36
Valves, H.P., one for each cylinder, diameter, inches	16½
I.P., two for each cylinder, inches	18½
L.P., two for each cylinder, inches	25
Travel of valve, H.P., inches	9
I.P., inches	9
F.L.P., inches	9⅝
A.L.P., inches	9⅝
Connecting rod, length between centers, inches	72
ratio to crank	4 to 1

Crank shaft, number of sections	2
diameter, inches	12¾
length, feet and inches, starboard	13-08¼
port	13-01½
Crank pin, diameter, inches	13½
length, inches	17½
Thrust shaft, diameter, inches	11½
length, feet and inches, starboard	20-00¾
port	14-06½
Line shaft, diameter, inches, starboard	11½
Stern-tube shaft, diameter, inches	12½
length, feet and inches	35-10¾
Propeller shaft, diameter, inches	12½
length, feet and inches	47-06¾

CONDENSERS.

Main Condensers.—There is one main condenser in each engine room, cylindrical in form, the inside diameter being 6 feet. The shell is $\frac{5}{16}$ inch thick, with three double-butt joints, with circumferential L-bar stiffeners. The water chests are cast composition ½ inch thick. The forward chest is the one for the entrance and exit of circulating water, and has a division plate fitted with valves which, when open, allow the circulating water to pass overboard direct, the valves being worked by a lever on the outside of the condenser. It has nozzles for inlet and outlet of circulating water 16½ inches in diameter. There are 4,085 seamless-drawn tubes in each condenser, $\frac{5}{8}$ inch outside diameter, No. 16 B.W.G. in thickness. The tubes are 12 feet long between tube sheets, and are spaced $1\frac{5}{16}$ inch between centers. Baffle plates are fitted to direct the steam over all the tubes. The cooling surface for each condenser is about 8,000 square feet, measured on the outside of the tubes. The material of the condenser is as follows: Shells, steel, class B boiler plate; baffle plates, steel, class C boiler plate; tubes, composition—copper 70, tin 1, zinc 29 per cent.; tube sheets and supporting plates are of rolled Naval bronze; glands are of tubing of the same composition as the tubes; water chests are of composition; all plates are of bronze.

Auxiliary and Dynamo Condensers.—In the forward engine room there is an auxiliary condenser connected

through the auxiliary exhaust pipe to all the auxiliary machinery. The condenser has an air and a circulating pump.

In the dynamo room there is an auxiliary condenser for the exclusive use of the dynamo engines. It has an air and a circulating pump.

The shells of all condensers are ¼ inch thick; the tube sheets are 1 inch thick.

The diameter and the spacing of the tubes and the packing is the same as in the main condensers.

EVAPORATING AND DISTILLING PLANT.

There are four evaporators and four distillers with their accessories. The evaporators have a combined capacity of 16,000 gallons of water per twenty-four hours, and the distillers of 16,000 gallons of potable water per twenty-four hours.

The evaporators take steam from the auxiliary steam pipe, and the coil drain pipes lead through by-pass automatic traps to feed tanks. The shells of the evaporators have connections with valves and pipes of approved size, for directing the steam into the distillers and into the auxiliary exhaust pipe.

The feed water for the evaporators is taken from the circulating pipe after it has passed from distillers and from the sea.

There are blow pipes of ample size so arranged that the brine in the evaporators may be blown overboard.

The distiller circulating pumps discharge to the distillers, the water passing overboard. There is also a direct connection from these pumps to the sanitary pipe, and the relief valve for the distillers is so placed as to act for both the distillers and the sanitary pipe.

In addition to the pump connection for distiller circulating, provision is made for circulating water through the distillers.

The evaporators are horizontal, and each has 223 square feet of tube-heating surface. The tubes are 2 inches outside diameter and are without bends.

Each distiller has 80 square feet of tube-cooling surface. The tubes are straight, ⅝ inch outside diameter, thor-

oughly tinned on both sides, and expanded and sweated into tube sheets.

FEED-WATER HEATERS.

In each engine room there is a feed-water heater with all necessary fittings complete. The cooling surface for each heater is 600 square feet, measured on the outside of the tubes.

The heater is of the direct-flow type located on the discharge side of the main feed pumps. The heating agent is auxiliary exhaust steam.

The tubes are $\frac{5}{8}$ inch outside diameter, of No. 16 B.W.G.

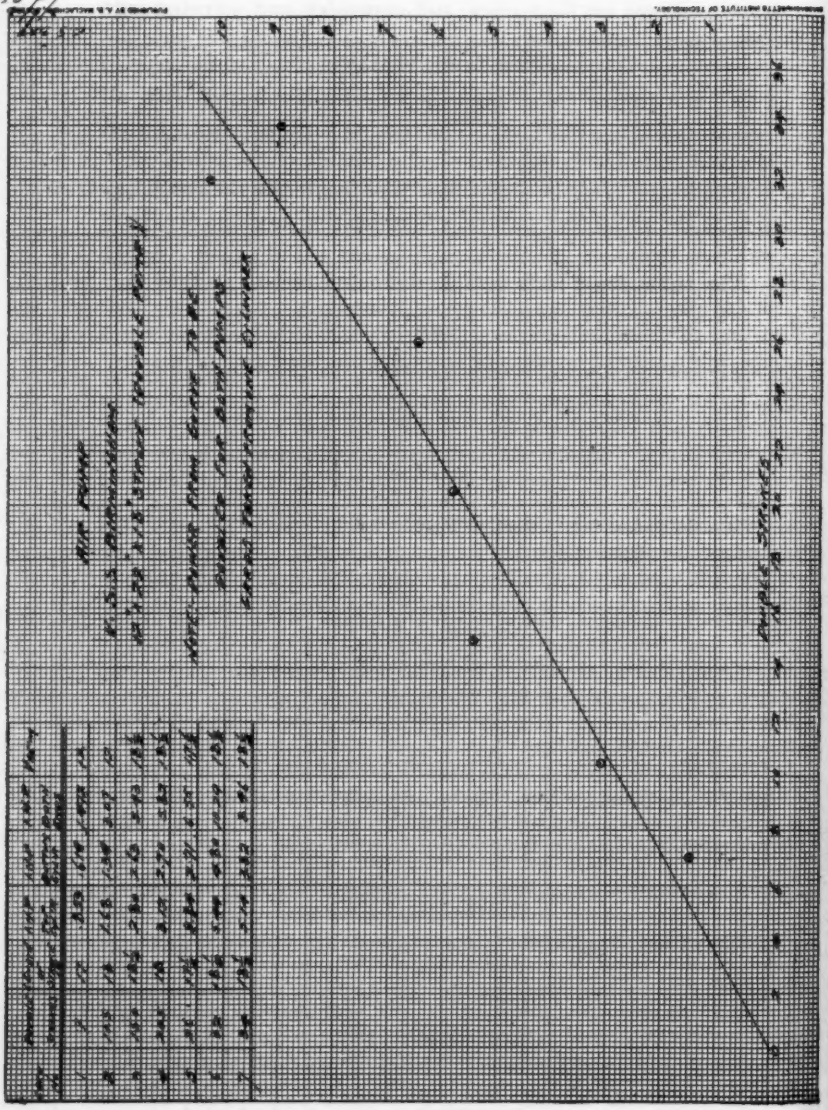
PUMPS.

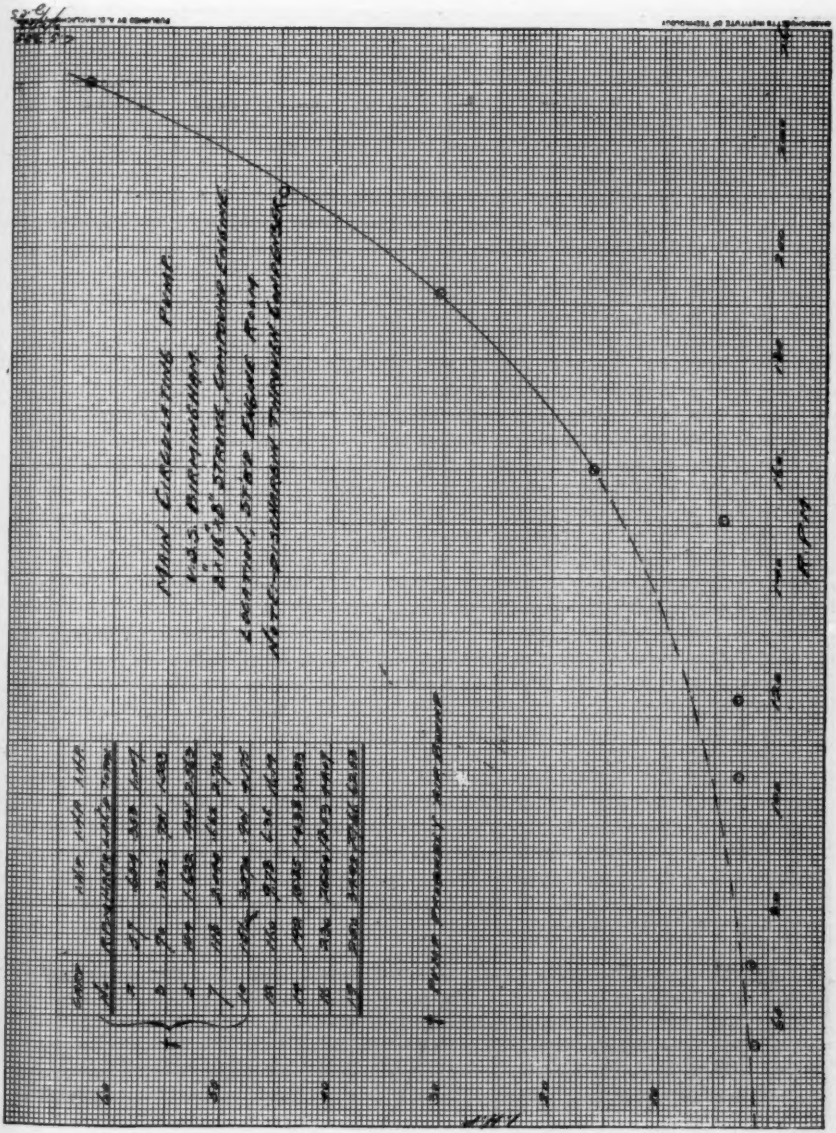
The pumps are in accordance with the following table:

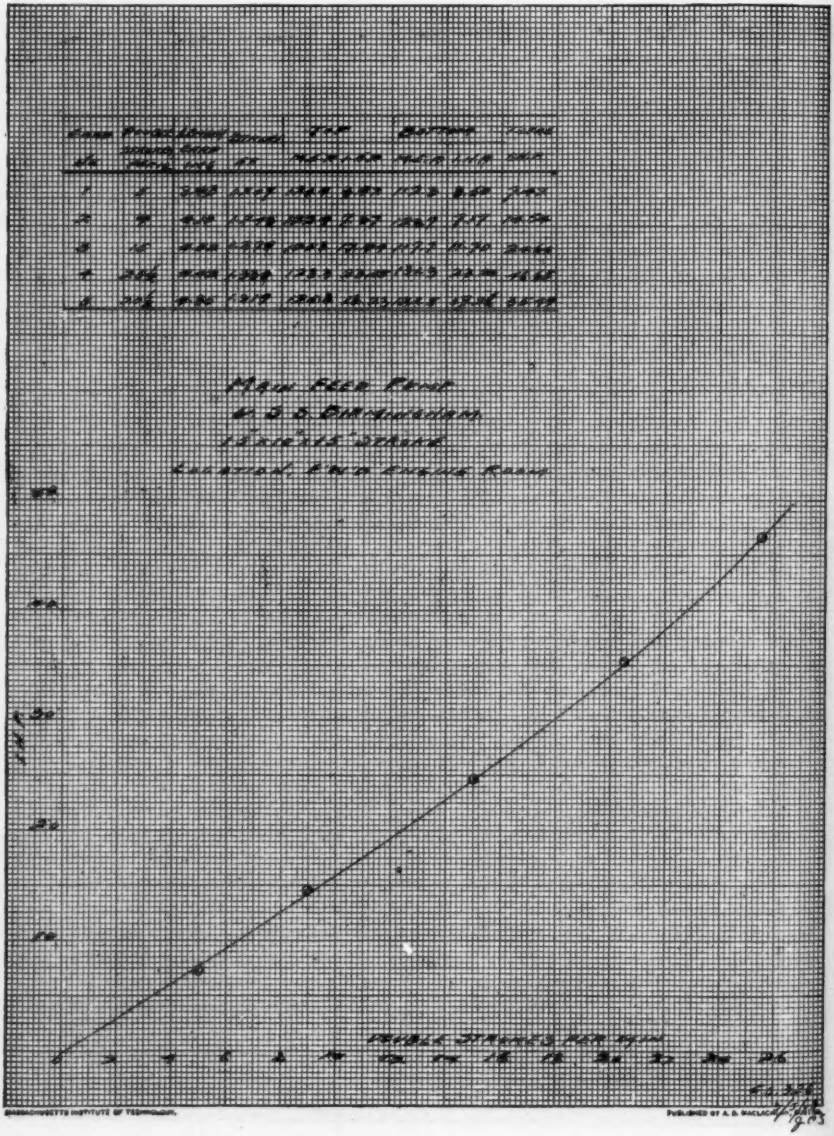
DATA OF PUMPS AND AUXILIARY ENGINES.

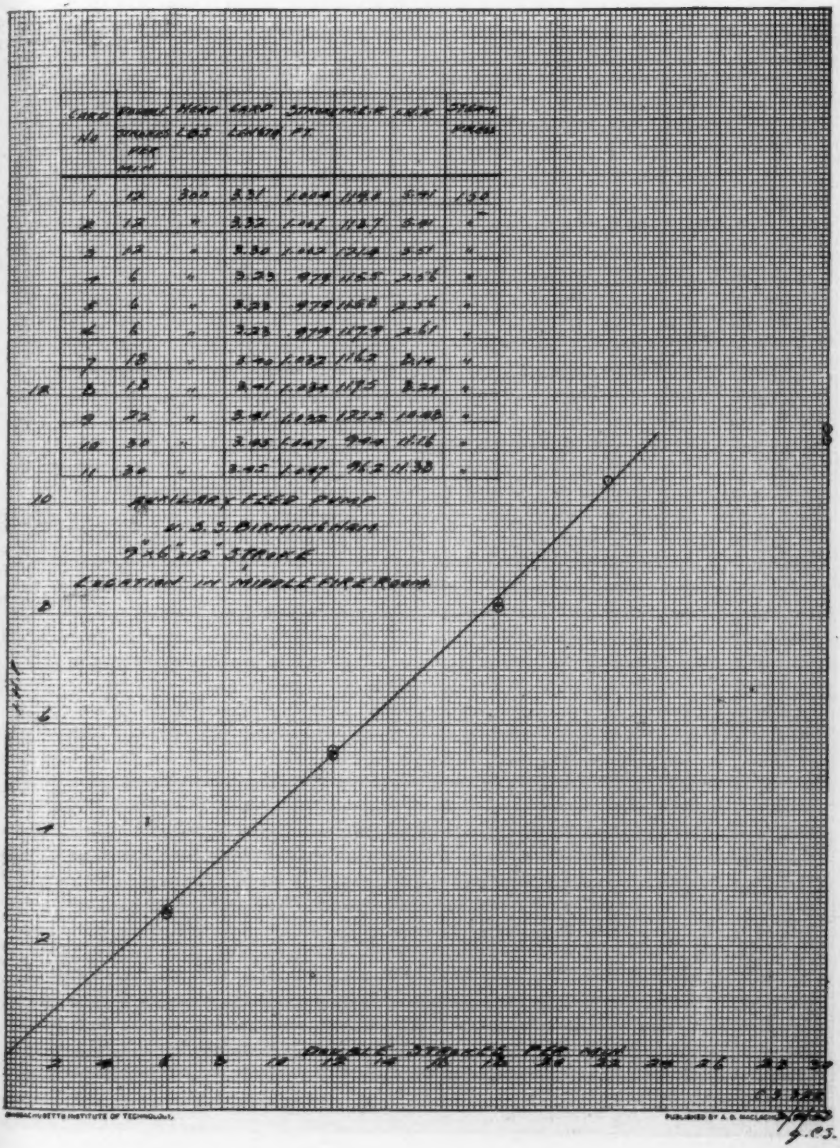
Auxiliary pumps.	Number.	Type, make and location.	Steam cylinders.				Water cylinders.			
			Number.	Diameter, inches.	Diameter piston rod, inches.	Stroke, inches.	Number.	Diameter, inches.	Diameter piston rod, inches.	Stroke, inches.
Main air	2	Blake vertical duplex, one in each engine room.	2	12	$2\frac{1}{2}$	18	2	22	$2\frac{1}{2}$	18
Main circulating	2	Centrifugal, driven by 8" and 15" × 8" vert. com. engine, one in each engine room.	2	8 & 16	$1\frac{3}{4}$	8	..	...	...	..
Main feed	2	One Blake vertical simplex in each engine room.	1	15	$2\frac{1}{2}$	15	1	10	$2\frac{1}{2}$	15
Auxiliary feed	3	One Blake vertical simplex in each fireroom.	1	9	$1\frac{3}{4}$	12	1	6	$1\frac{3}{4}$	12
Engine-room fire and bilge	2	One Blake vertical simplex in each engine room.	1	10	$1\frac{3}{4}$	12	1	$8\frac{1}{2}$	$1\frac{3}{4}$	12
Reserve feed	1	One Blake vertical simplex in forward engine room.	1	$3\frac{1}{2}$	$\frac{3}{4}$	4	1	4	$\frac{3}{4}$	4
Auxiliary condenser	1	Blake horizontal simplex combined air and circulator in forward engine room.	1	6	$1\frac{1}{2}$	12 Air.	1 1	10 10	$1\frac{1}{2}$ $1\frac{1}{2}$	12 12
Dynamo condenser	1	Blake horizontal simplex combined air and circulator in dynamo room.	1	6	$1\frac{1}{2}$	12 Air.	1 1	8 8	$1\frac{1}{2}$ $1\frac{1}{2}$	12 12
*Evaporating and distilling plant.	2	Dist. circ. horiz. simp., Orlop deck.	1	10	$1\frac{1}{2}$	12	1	10	$1\frac{1}{2}$	12
	1	Evap. feed vert. simp., Orlop deck.	1	$4\frac{1}{2}$	$\frac{1}{2}\frac{3}{8}$	6	1	5	$\frac{1}{2}\frac{3}{8}$	6
	1	Fresh-water vert. simp., Orlop deck.	1	$4\frac{1}{2}$	$\frac{1}{2}\frac{3}{8}$	6	1	5	$\frac{1}{2}\frac{3}{8}$	6
Fireroom fire and bilge	3	One Blake vertical simplex in each fireroom.	1	12	2	12	1	8	2	12

* Maximum rated output of fresh water (distilled) per 24 hours, 16,000 gallons.









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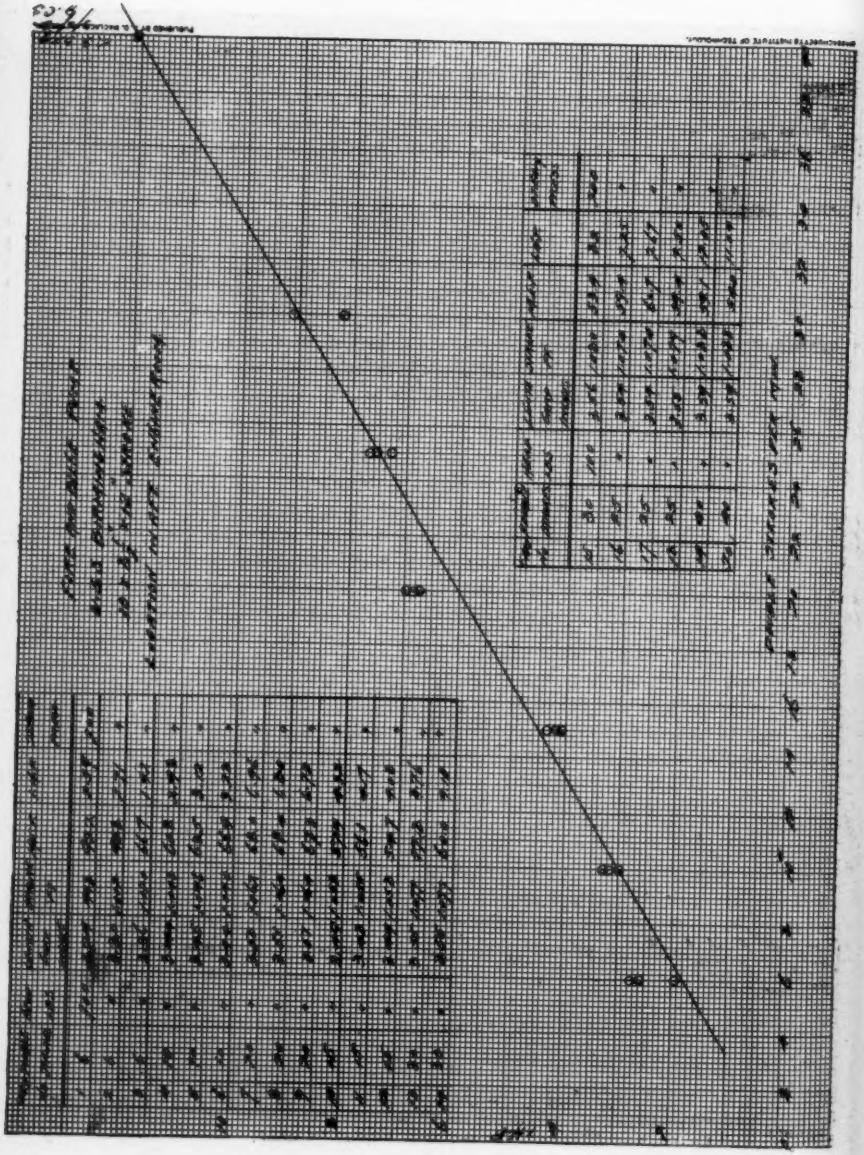
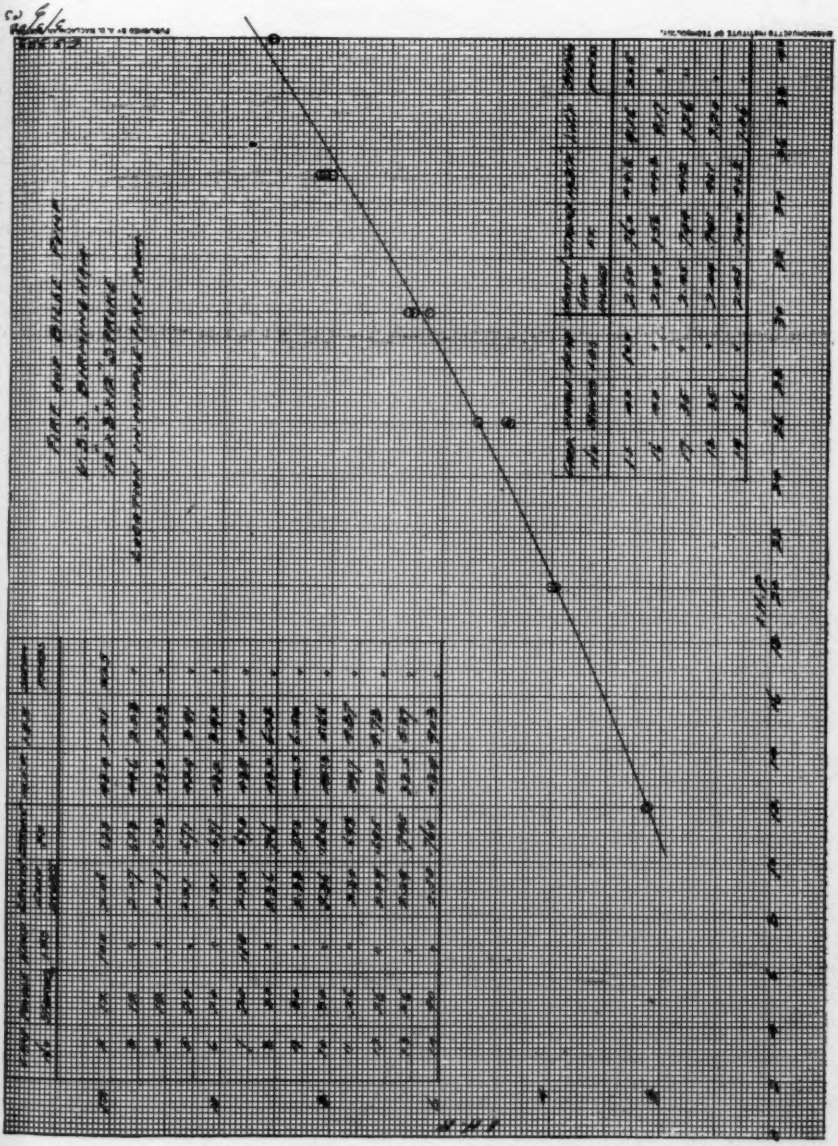
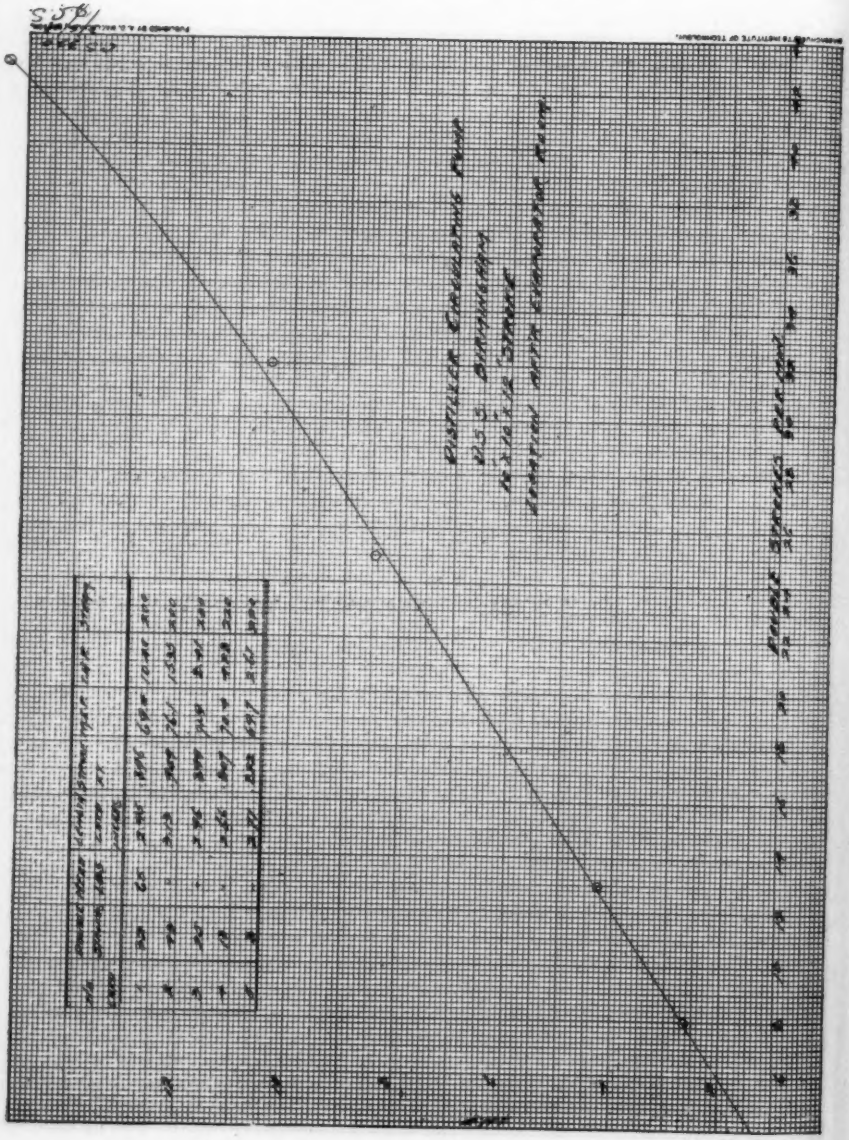
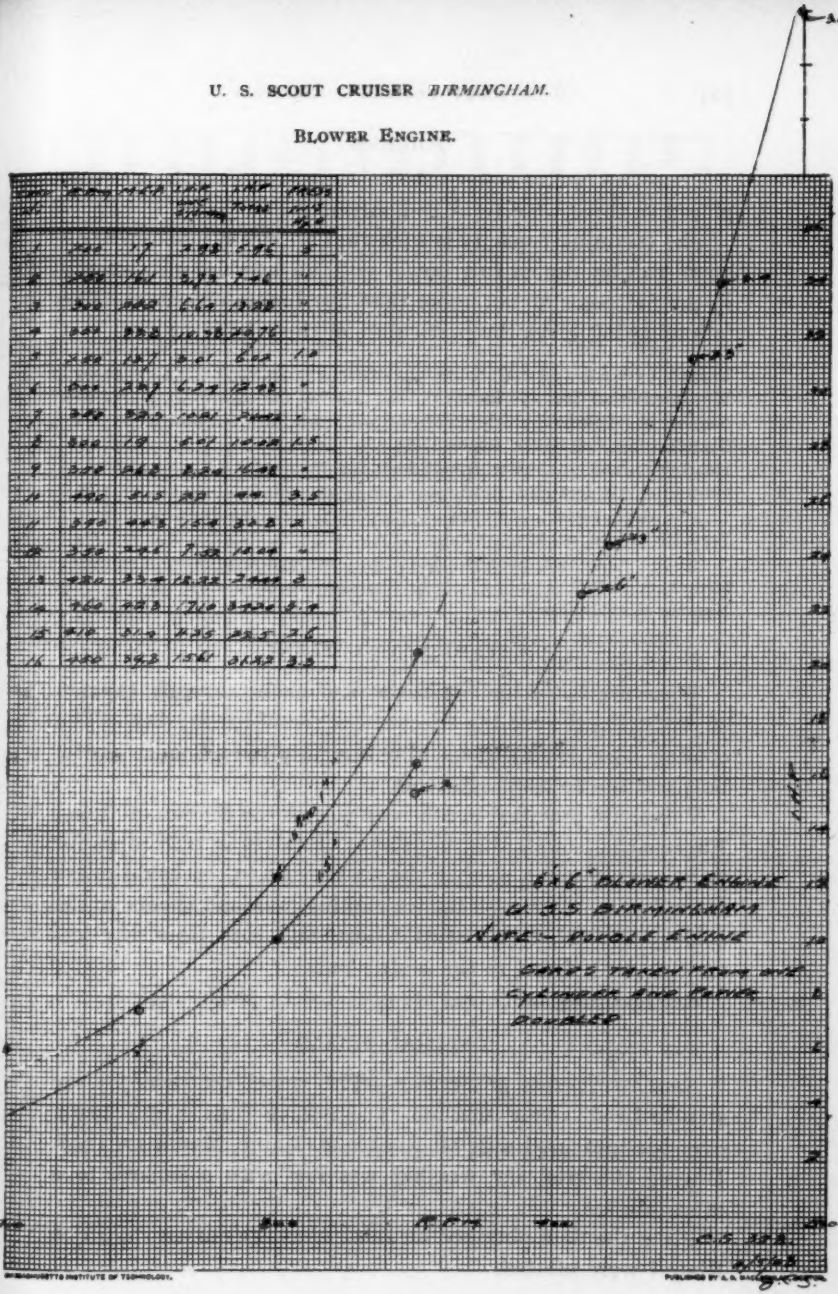






U. S. SCOUT CRUISER *BIRMINGHAM*.

BLOWER ENGINE.



BOILERS.

There are twelve water-tube boilers of the Fore River type, placed in three watertight compartments.

The steaming capacity is such that all steam machinery on board can be run at full power with an average air pressure in the firerooms of not more than five inches of water. The following list gives the particulars of these boilers:

Length, external, feet and inches	11–02
Width, external, feet and inches	12–00½
Number of furnaces	1
Grate surface, one boiler, square feet	58
Heating surface, one boiler, square feet	3,166
Grates, length, feet	7.25
width, feet	8
Pressure, design, working, pounds	275
test, pounds	400
Ratio G.S. to H.S	54.6
Number of tubes, one boiler	1,462

REFRIGERATING PLANT.

There are two Allen dense-air ice machines, each capable of producing the cooling effect of one ton of ice per day. The cooling pipes from the machine lead into the ice tank, scuttle butt and the cold-storage room.

Valves are provided in accordance with the Bureau of Steam Engineering standard arrangement of valves in cold pipes for refrigerating plant, so that the air may go to the cold-storage room direct, or through the ice-making tank and thence to the cold-storage room and scuttle butt, and also from the ice-making tank direct to the scuttle butt.

Refrigerating machines are operated by steam engines.

OFFICIAL TRIALS.

Standardization of Screws.—The vessel was tried by the standardized screw method, progressive runs being made over the measured mile off Rockland, Maine, March 11, 1908. During the trial the weather was fair with a smooth to a moderate sea.

From the data obtained on these runs the curves shown on Plate I were plotted.

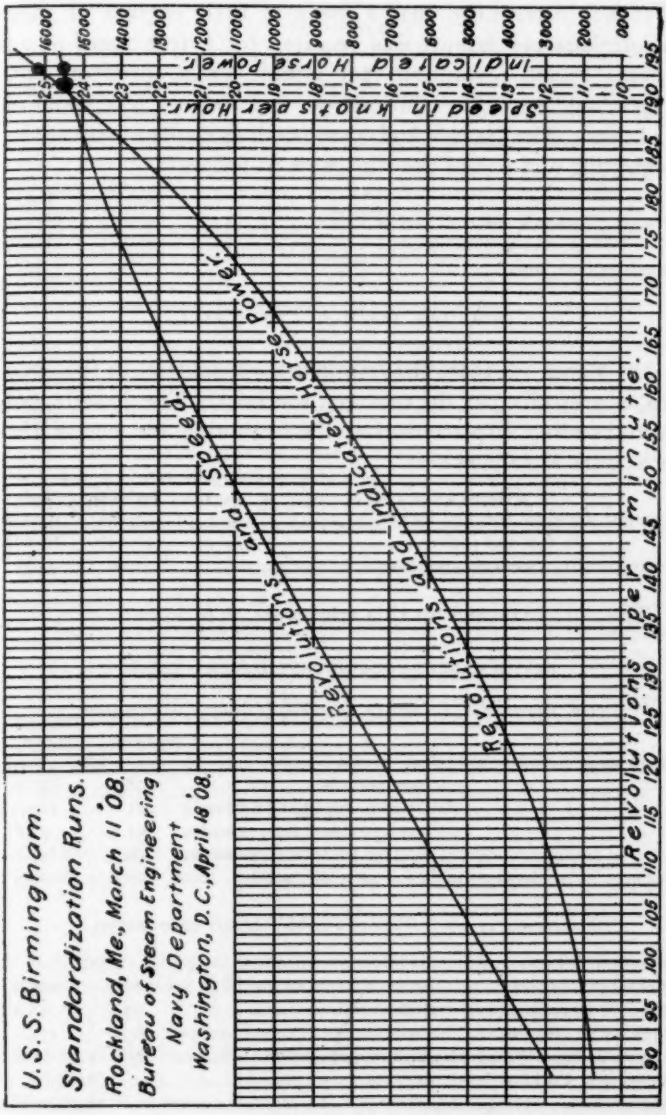

U.S.S. Birmingham.
Standardization Runs.
Rockland, Me., March 11 '08.
Bureau of Steam Engineering
Navy Department
Washington, D.C., April 18 '08.
Revolutions per minute.
Revolutions - Indicated Horse-power.
Revolutions - Speed.
Speed in knots per hour.
Indicated Horse Power.

Plate I.

It was determined that a mean of both engines of 89.7 revolutions per minute was required for a true speed of 12 knots; 170.38 revolutions per minute for a speed of 22.5 knots; and 187.23 revolutions per minute for a true speed of 24 knots. The draught and corresponding displacement at the beginning and end of the runs were as follows:

	Beginning.	End.
Draught, forward, feet and inches	16-02⅝	15-11½
aft, feet and inches	17-02¼	17-01⅝
Displacement, tons	3,757	3,701

OFFICIAL FOUR-HOURS' TRIAL.

On March 12, 1908, the four-hours' official trial in the open sea, as prescribed by the contract, was held. The draught and corresponding displacement at the beginning and end of the runs were as follows:

	Beginning.	End.
Draught, forward, feet and inches	16-08¼	16-03¾
aft, feet and inches	16-09¼	16-08⅛
Corresponding displacement, tons	3,768	3,676

The data obtained on this trial were as follows:

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	250	
engines, pounds	238.0	230.0
H.P. steam chest gauge, pounds	221.0	229.0
1st receiver (absolute), pounds	92.4	97.6
2d receiver (absolute), pounds	26.3	27.1
Vacuum in condensers, inches of mercury, mean	26.6	26.3

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	37	
Discharge, degrees	104.0	105.0
Hotwell, degrees	99.0	98.0
Feed water, degrees	185.0	174.0
Engine room, working platform, degrees	83.0	78.0
Firerooms, working level, degrees	81.0	
Smoke stacks, average, degrees	665.0	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Average revolutions, main engines, per minute	192.96	190.35
Pumps, main air	22.7	21.4
circulating	180.0	182.0
feed, d.s., per minute	38.4	31.0
fire and bilge	...	25.0
Dynamos, engines	400	
Blower engines	430	
Speed of ship, in knots per hour	24.326	
Slip of propeller, in per cent. of its own speed, based on mean pitch	16.40	15.23
Air pressure in firerooms, in inches of water, mean	2.54	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	109.4	107.7
I.P. cylinder	46.1	47.1
F.L.P. cylinder	11.5	11.9
A.L.P. cylinder	17.4	13.7
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	37.5	36.2

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	2,349.0	2,285.0
I.P. cylinder	2,551.0	2,571.0
F.L.P. cylinder	1,212.0	1,241.0
A.L.P. cylinder	1,833.0	1,434.0
total	7,945.0	7,531.0
Collective H.P. of both main engines	15,476.0	
Main air, circulating and feed pumps	194.0	
Other auxiliaries	219.0	
Total	15,889.0	

COAL.

Kind and quality used on trial.....Pocahontas, run of mine.
Pounds, per hour, main and auxiliary engines, during trial..... 2,990.4

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	22.83
Pounds of coal per I.H.P. per hour, main engines	1.932
all machinery in operation	1.882
square foot of grate surface per hour	42.96
Cooling surface (main condenser), square feet per I.H.P. (main engines)	1.032
Heating surface, square feet per I.H.P. (total)	2.39

OFFICIAL TWENTY-FOUR HOURS' TRIAL.

On March 12, 1908, the official twenty-four hours' trial at 12 knots was commenced. The weather during this trial was fair the first part but became overcast and threatening during the latter part. The wind was very moderate to stiff breezes from the N. to S.S.E. and S. The sea was moderate.

The data obtained on this trial were as follows:

PERFORMANCE.—TWENTY-FOUR HOURS' OFFICIAL TRIAL AT TWELVE KNOTS.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	206.0	
engines, pounds	205.0	199.0
H.P. steam-chest gauge, pounds	49.0	54.0
1st receiver (absolute), pounds	19.0	24.0
2d receiver (absolute), pounds	5.80	7.60
Vacuum in condensers, inches of mercury, mean	26.4	27.1

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	38.2	39.3
Discharge, degrees	50.0	62.3
Hotwell, degrees	102.0	...
Feed water, degrees	183.0	...
Engine room, working platform, degrees	75.5	85.1
Smoke stacks, average, degrees	352.0	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per minute	91.40	91.26
Pumps, main air	17.4	16.5
circulating	151.6	109.0
feed, d.s., per minute	15.0	...
Dynamos	400.0	
Blower engines	262.0	
Speed of ship, in knots per hour	12.225	
Slip of propeller, in per cent. of its own speed, based on mean pitch	11.30	11.23
Air pressure in firerooms, in inches of water, mean	0.25	

Mean Effective Pressures in Cylinders, in pounds per square inch. (Averages of cards taken at half-hourly periods.)

	Starboard.	*Port.*
Main engines, H.P. cylinder	34.0	35.8
I.P. cylinder	8.6	12.2
F.L.P. cylinder	1.48	1.4
A.L.P. cylinder	2.01	2.03
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	7.61	8.58

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	344.0	365.0
I.P. cylinder	226.0	317.0
F.L.P. cylinder	74.0	69.0
A.L.P. cylinder	100.0	101.0
total	744.0	852.0
Collective H.P. of both main engines	1,596.0	
main air, circulating and feed pumps	54.0	
Other auxiliaries	103.0	
Total	1,753.0	

COAL.

Kind and quality used on trial	Pocahontas, run of mine.
Pounds, per hour, main and auxiliary engines during trial	4,629.0

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	7.56
Pounds of coal per I.H.P. per hour, main engines	2.90
all machinery in operation	2.64
square foot of grate surface, per hour	19.95
Cooling surface (main condenser), square feet per I.H.P. (main engines)	10.01
Heating surface, square feet per I.H.P. (total)	7.22

OFFICIAL TWENTY-FOUR HOURS' TRIAL, AT TWENTY-TWO AND ONE-HALF KNOTS PER HOUR.

On March 14, 1908, the twenty-four hours' official trial at 22½ knots per hour, according to contract, was commenced. The weather during this trial was at first fair, but foggy and overcast during the latter part. The wind was at first light, but gradually increased to a stiff breeze, developing a moderate sea and swell.

The data obtained on this trial were as follows:

PERFORMANCE.—TWENTY-FOUR HOURS' OFFICIAL TRIAL AT TWENTY-TWO AND ONE-HALF KNOTS.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	203.0	
engines, pounds	192.0	181.0
H.P. steam chest gauge, pounds	171.0	178.0
1st receiver (absolute), pounds	67.6	77.2
2d receiver (absolute), pounds	20.0	22.6
Vacuum in condensers, inches of mercury, mean	27.0	27.1

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	39.5	...
Discharge, degrees	102.0	105.0
Hotwell, degrees	99.0	...
Feed water, degrees	177.0	168.0
Engine room, working platform, degrees	78.2	...
Smokestacks, average, degrees	554.0	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per minute	172.2	172.0
Pumps, main air	23.1	21.2
circulating	160.6	145.4
feed, d.s., per minute	30.0	30.0
fire and bilge	...	19.5
Dynamos	400.0	
Blower engines	404.0	
Speed of ship, in knots per hour	22.668	
Slip of propeller, in per cent. of its own speed, based on mean pitch	12.71	12.59
Air pressure in firerooms, in inches of water, mean	1.98	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	90.3	89.9
I.P. cylinder	35.2	37.8
F.L.P. cylinder	8.61	9.3
A.L.P. cylinder	9.78	10.72
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	27.69	29.09

INDICATED HORSEPOWER.

	Starboard.	*Port.*
Main engines, H.P. cylinder	1,743.0	1,739.0
I.P. cylinder	1,748.0	1,876.0
F.L.P. cylinder	814.0	878.0
A.L.P. cylinder	937.0	1,016.0
total	5,242.0	5,509.0
Collective H.P. of both main engines	10,751.0	
main air, circulating and feed pumps	166.0	
Other auxiliaries	181.0	
Total	11,098.0	

COAL.

Kind and quality used on trial.....Pocahontas, run of mine.
Pounds, per hour, main and auxiliary engines, during trial..... 20,510.0

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	15.945
Pounds of coal per I.H.P. per hour, main engines	1.908
all machinery in operation	1.848
square foot of grate surface, per hour	29.47
Cooling surface (main condenser), square feet per I.H.P. (main engines)	1.485
Heating surface, square feet per I.H.P. (total)	3.423

SERVICE TEST ON THE STEAMSHIP *GOVERNOR COBB.**

(PARSONS TURBINE.)

By W. S. Leland and H. A. Everett.

The test on the steamship *Governor Cobb*, of the Eastern Steamship Company, was run by the Massachusetts Institute of Technology to provide thesis work for some of the graduates in the Department of Naval Architecture.

The test was run under service conditions on the regular trip of the *Cobb* from Boston to St. Johns, *via* Portland and Lubec. Observations began on passing Boston Light and were continued for twenty-six hours—for the boiler test continuously; for the engine test at favorable times.

All observations were plotted, which presents an interesting study of the results and makes simultaneous readings possible with a limited corps of observers. The close agreement of these curves is a good check on the accuracy of the observations.

Time has not permitted the preparation of a complete report, but merely a summary of that portion of the test run at full speed under the most favorable conditions.

The horsepower was determined by means of the Denny and Johnson torsion meter belonging to the United States steamship *Chester*, which was loaned by the Bureau of Steam Engineering, United States Navy Department, through the courtesy of Admiral Charles W. Rae, Chief of the Bureau. The details of the loan were arranged by Benjamin C. Bryan, Lieutenant Commander, U. S. N., who gave the matter his personal attention and accompanied our party on the test.

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* Reprinted from "Transactions of the Society of Naval Architects and Marine Engineers," 1907, 15, 117–120, plates 37–39. Copyrighted.

The torsion meter was set up in the engineering laboratory and a thorough working knowledge obtained by the use of experimental apparatus before installing the meter on board. Thirty-six feet on the side shafts, and 49 on the center shaft between inductors, was the greatest length obtainable, which gave a meter reading of about .50 and .73, respectively, at full power.

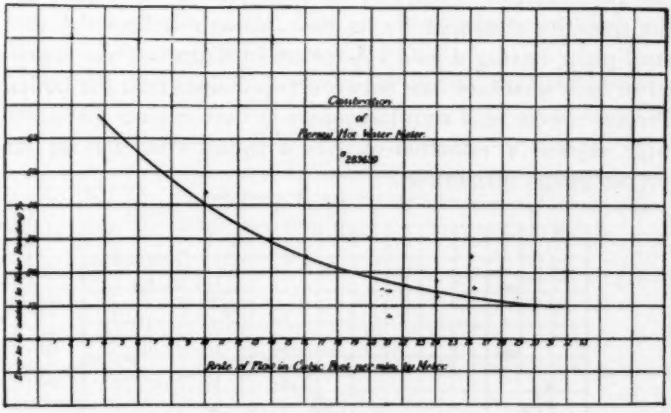

FIG. 1

In computing the horsepower, 1.506, based on an assumed torsional modulus of elasticity of 11,600,000, was used for the constant K in the formula:

$$\text{H.P.} = \frac{Kd^3rR}{CL},$$

in which d = diameter of shaft in inches ($6\frac{3}{8}$).

r = torsion meter reading.

R = revolutions per minute.

C = inductor constant (12.5).

L = length of shaft in feet between inductors.

The water consumption was measured by a Hersey hot-water meter, loaned by the Hersey Manufacturing Company, of South Boston. It was installed in the suction line between the hot well and the feed pump, and gave exceedingly satisfactory results.

This meter was later calibrated under similar conditions, the curve in Figure 1 showing the percentages of error. The plot of meter readings was struck in as a straight line, showing practically a uniform rate of water consumption, no point varying from the line by a quantity greater than 1 per cent. of the total.

The steam for all auxiliary purposes was passed through the two auxiliary lines, one on each side of the vessel, and the quantity measured by its flow through orifices. A thin steel plate, having a hole $1\frac{5}{16}$ inches in diameter, was inserted in each auxiliary line between two flanges near the boiler. Pressures were read simultaneously at both orifices and at no time showed a variation of over a pound after making the proper gauge corrections.

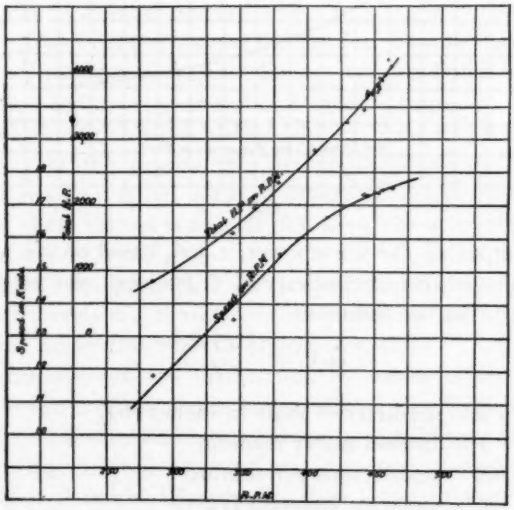

FIG. 2

The orifice was afterwards set up in the laboratory and its coefficient carefully determined by actually weighing the condensed steam under conditions similar to those on the boat.

Several buckets of coal were weighed and their average,

which varied only ten pounds from either maximum or minimum, multiplied by the number of buckets, was taken as the coal consumption. The plot of coal consumption, like that of the water, is a perfectly straight line, showing a uniform rate.

The run from Boston to Portland was largely consumed in a progressive trial, the speed being taken by a stop-watch and a McGray electric log towed from the end of a boom well clear of the wake. The log had previously been calibrated by towing over the measured mile.

Results of speed and power are shown in the curves in Figure 2.

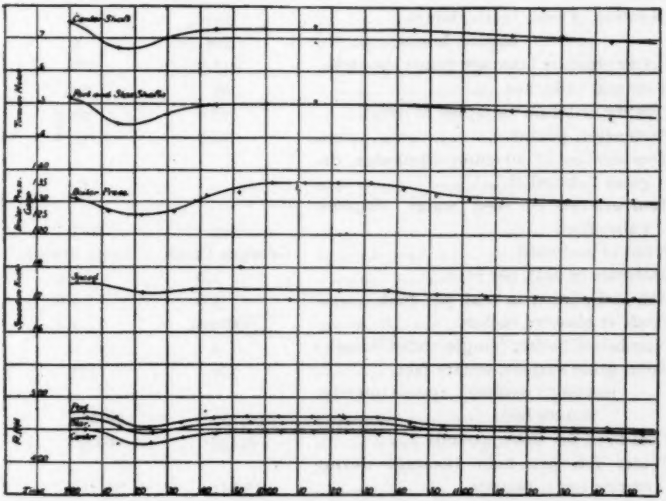

FIG. 3

The best run was made at full speed between Portland and Lubec, under the most favorable conditions of weather and sea. All observations were taken at ten-minute intervals excepting the coal, which was recorded every fifteen minutes. Figure 3 shows the chief records during this test.

An attempt was made to determine the quality of steam, but as there were objections to tapping the main pipe a sample was taken from the drip connection, which showed 2.5 per cent. of moisture, which is more than would be obtained from a fair sample. It may be of interest to compare the following tabulated results with similar results obtained from a test on the steamship *Nantucket*, of the Merchants' and Miners' Transportation Company.

RESULTS OF TEST.

	Nantucket.	*Cobb.*
Type of engine	Reciprocating.	Parsons Turbine.
Date of test	February 7, 1904.	April 18, 1907.
Duration of test, boiler, hours	20½	8
engine, hours	20½	4
Boiler pressure (average gauge) pounds	147.3	128
Vacuum, in inches	25	27
Quality of steam (sampled at drip)	98.8	97.3
Barometer, pounds	14.7	14.7
Temperature of air-pump discharge, degrees Fahrenheit	...	110
Temperature of feed water, degrees, Fahrenheit	209.4	213
Kind of coal used	Georges Creek.	Cape Breton.
Moisture in coal, per cent	2.0	1.9
Ash and clinker in coal, per cent	7.6	6.8
Draft at blowers, inches	Natural.	2.1
Number of boilers (single-ended Scotch)	4	6
Total grate surface, square feet	320	323
heating surface, approximately, square feet	10,150	12,000
Coal fired per hour, pounds	5,135	8,050
Water fed per hour (average during engine test), pounds	45,844	85,710
Coal burned per square foot grate surface, pounds	16	24.64
Maximum revolutions, port	...	475
starboard	...	460
center	...	450
Corresponding total shaft horsepower	...	4,100
Average revolutions	74.05	447
total horsepower	2,362 I.H.P.	3,747 S.H.P.

	Nantucket.	*Cobb.*
Steam for auxiliaries, pounds	...	38,360
per I.H.P. per hour, total pounds	19.41	...
shaft horsepower per hour for all purposes	...	22.67
Steam per shaft horsepower per hour (propelling machinery only), pounds.	...	19.74
Speed (average), knots	15.00	17.21

DIMENSIONS.

	Nantucket.	*Cobb.*
Length between perpendiculars, feet	274	290
Beam molded, feet	42	51
Draught, feet	15	13
Displacement, tons	2,700	...

NOTES.

THE WEIGHT OF MARINE TURBINES.

That the weight of marine turbine machinery has been greatly increased during the last two or three years is well known, and as the tendency is still towards increase without any proportionate gain in economy, it may be as well to examine the causes which have led up to this, and indicate some means whereby it can advantageously and safely be reduced. To some extent a considerable increase was necessary on account of the absolute unsuitability of the original marine installations to the commercial necessities of marine engineering. For instance, in many vessels the replacing of a broken gland ring, involving a few minutes' work in a reciprocating job, necessitated lifting the cylinder cover and rotor from the turbine. A large increase in weight of astern turbines was also necessary, the early turbine vessels being lamentably deficient in maneuvering power. More rigid cylinder and rotor construction was also desirable, and in the case of the new Cunarders and the ocean-going destroyers, it was found necessary greatly to increase the blade sections, as the stress in these, due more especially to end thrust, was exceptionally high. All this involved increase of weight for the same power and speed of rotation, and developments in propeller design involving greatly reduced revolutions added still more. For the same efficiency, the weight of a turbine of a given power will vary inversely as the square of the revolutions, so that a reduction from 500 to 400 revolutions per minute, brought about by the more advantageous use of a larger propeller, involves an increased weight of over 60 per cent. The use of much heavier cylinders than Parsons originally tried was due to substitution of cast iron for cast steel. Not only

were cylinders extremely hard to obtain in the latter metal—the *Amethyst* had numerous failures before a good one was secured—but cast steel has an awkward habit of warping, and distortion of cylinders has accounted for endless blade stripping. In warships the use of cruising turbines has greatly augmented the weight per horsepower. Lagging, thrust-adjustment gear, oil-cooling and pumping systems, outside glands with mazes of valves and pipes for a most complicated steam-packing system, are all items that have been enormously increased without attracting much notice. It is very simple to advocate reduction, but the fact is that in the early stages the marine turbine, whatever future promise it had in store in the eyes of those that were first acquainted with it, was a crude and undeveloped machine that needed a great deal of modification before it was made commercially suitable. That modification meant increasing the weight.

The following table gives the weights of turbine machinery only, excluding all auxiliaries, condensers, or shafting for various classes of vessel:

WEIGHT OF TURBINES PER HORSEPOWER.

Type of ship.	Shaft H.P.	Revolutions.	Weight of turbine, tons.	Weight per H.P., pounds.
Battleship.....	25,000	320	450	40.4
Ditto, proposed.....	28,000	275	540	43.2
Small cruiser.....	14,000	500	180	28.8
Torpedo-boat destroyer	18,000	700	80	10.0
Ditto..	4,000	1,100	18	10.1
Channel steamer.....	12,000	500	110	20.5
Atlantic liner (type)...	42,000	250	900	48.0
Intermediate steamer...	10,000	350	180	40.25

That the present weight of machinery can be greatly reduced there can be no doubt whatever, but it involves a very much clearer appreciation of the functions and performance of the numerous details in an installation than now appears to be possessed in some quarters.

Commencing with the turbine cylinder, in nearly all cases do we find circumferential ribs of some kind; either the half-round rib, typical perhaps of East Coast practice, or the heavy

bulb rib that has distinguished Clyde cylinders for years. A turbine cylinder after all is a pipe in the position of a beam fixed at the after end and free to slide (supported) at the forward end. When working it is under gradually decreasing internal pressure. Who would rib a pipe, and if so, what use are bulb ribs if half-round ones do, as has been amply proved? What use, in any case, are the shallow half-round ribs? This, at least, is a point on which practice is not only not consistent, but not apparently correct in its policy from the beginning.

Then, again, take the test pressure enforced. Without mentioning names, we can quote instances of Admiralties or registration societies, acting doubtless on the recommendations of their technical advisers, who insist that a high-pressure cylinder shall be tested to one and a-half times the boiler pressure, and that the forward end of the low-pressure cylinder shall be tested to one-fifth the boiler pressure. "Escape valves will be fitted at the after end of the high-pressure cylinders," says the specification. Now, with a boiler pressure of 250 pounds this means that the cylinder end and the last expansion of the high-pressure have to withstand 375 pounds per square inch, while the end of the low-pressure and its first expansion, which, when working, are dealing with an identical pressure to that in the end of the high-pressure, have only to withstand 50 pounds per square inch. Assuming this turbine to be working between 185 pounds per square inch and 28 inches vacuum, the maximum working pressure in the receiver pipe will only be between atmosphere and 5 pounds gauge; so that even without the escape valve the test pressure is enormously exaggerated. This is another direction in which weight can be safely reduced by proper appreciation of the function of the various parts. The two cylinders effect the expansion of the steam in six stages in each—twelve in all, or, say, the first four correspond to the high-pressure cylinder, the last two in the high-pressure and first two in the low-pressure corresponding to the intermediate cylinder of a reciprocating engine. Even in the days of 300 pounds pres-

sure and Belleville boilers, no one was found to require a water-pressure test of more than 200 pounds per square inch on an intermediate cylinder, or only about half what the much larger turbine barrel has to stand. These high test pressures, quite apart from adding to weight, are objectionable on the grounds of causing permanent deflection in the castings. It is consequently never possible to put this pressure on after the cylinders have been finished in the boring machine.

Many of the foremost marine engineers in the country are greatly against the system now so generally adopted by the Admiralty of making sets of engines for different ships interchangeable. Whatever its military advantages may be, there can be no doubt that it does not make for thoroughly careful design on the part of one firm when it knows that all its drawings will be given to another firm to work to, however amicably they may arrange the interchange. That basic plans should be got out by one company and adhered to "by command" can only restrict progressive design, and we are not surprised, therefore, when we find a fault made in one design carefully duplicated in several others. Unnecessary length between bearings, excessive scantlings in cylinder-end pedestals, thrust blocks or bearing covers, should all be subjected to far keener criticism from the manufacturers' point of view than is at present encouraged by the existing system.

Less remains to be done in the way of reducing the weight of rotors in existing designs than in the case of cylinders. The essential drum and end spindles will remain, but some alteration will probably take place in the form of wheel adopted for connecting the two. A cast-steel hollow-spoked spindle has up till recently been adopted, this form having been calculated as being best adapted to resist compression and twisting on a given weight of material. The recent trouble, however, with the cast-steel rotor wheels of H. M. S. *Superb* has resulted in an order being placed for this item in forged steel, though with practically no difference to the weight.

Although somewhat outside the weight of turbines proper, the condensing plant forms no inconsiderable portion of the total weight. The design of marine condensers at present reflects little credit on the engineers responsible, some of the condensers in the destroyers being nothing beyond circular or oval boxes of tubes. Less water and pumping power and much smaller condensers will result when the—somewhat obvious—fact is grasped that an efficient square foot is more use than many square feet of not only useless but harmful surface. A great deal remains to be done in leading the steam properly among the tubes, and steam speeds and passages need much revision. When this is done, and the surface in a condenser made as equal in efficiency as possible all over, the size and weight of condensing plants will also be much reduced. The arrangement of exhaust pipes to the condenser in most marine installations is not one that is well adapted for distributing the steam properly over the tubes. Adequate area is seldom or never allowed through the tube rows nearest the inlet, and much greater condensing efficiency must obviously result from a more equable distribution of work throughout the condenser.

The size of steam piping and valves merits attention with turbine machinery, the permissible speeds of steam and exhaust being much higher, thereby admitting of reduced pipe diameters. The fall of pressure between the throttle valve and turbine is constant, whereas in a reciprocating engine the closing of the slide valve induced a pulsating effect that demanded a slow speed of delivery. A very slight pressure drop is sufficient to cause a steam speed far in excess of that academically laid down in text books for the diameter of steam pipes.

Many other points might be cited, but in dealing with the above we have been careful to bear in mind specific cases of exaggerated dimensions or unsuitable proportions in recent practice. What we now desire to call attention to has reference to future practice more particularly, though the matter has for long enough been fully realized in certain specialist circles.

From the point of view of weight, cost and efficiency, the determination of the best dimensions of the marine turbine depends on the accuracy with which it is proportioned to the work it has to perform, and on the correctness with which the balance between the turbine and the propeller is struck. We have numerous cases in which the propeller is too large, or rather larger than is advisable for the turbine, but far more frequently do we find an unnecessarily powerful turbine. Setting aside for the moment the determination of relative proportion, the first step must at least be to determine propeller dimensions for any given power, speed and number of screws. A simple calculation shows the difference that may be made by the adoption of very slightly different ratios and pressures in the same type of design. Take a Channel steamer of 24 knots speed and 12,000 shaft horsepower. The propeller efficiency is known to be about 50 per cent. There will be three shafts.

Effective horsepower $=12{,}000\times.5=6{,}000$,

Effective thrust $=\frac{6{,}000\times 326}{24\times .95}$ where .95 is the wake factor

$=85{,}700$ pounds.

Some designers will allow a thrust pressure per square inch projecting area of 11 pounds in association with a projected surface ratio of 0.45, while others would go to 11.5 and a surface ratio of 0.5. The resulting diameters would be 7 feet $1\frac{1}{2}$ inches and 6 feet $4\frac{1}{2}$ inches respectively, and, with a pitch ratio of 0.95 and 20 per cent. slips in each case, the revolutions would be about 450 and 500 per minute respectively. The greatest care must be taken to use pressures and proportions that admit of efficient high speeds, as in this case one turbine set will weigh approximately 15 to 20 per cent. more than the other.

Equally important is the accurate determination of effective horsepower necessary to propel a given ship at a given speed. There are innumerable instances of this being exaggerated in the design, and the vessel attaining an unlooked for speed. One of the latest cases is undoubtedly that of the United

States cruiser *Chester*, which was designed for 24 knots and obtained 26.52, an increase in speed corresponding to about 33 per cent. in power. Another case is that of the *Tartar*, which was designed for about 15,000 shaft horsepower and gave 20,000. To some extent the boilers may have been too generously proportioned, but in each case the turbine was either able to pass over 30 per cent. more steam than it was designed for, or else, for its designed power, it was unnecessarily large. In the above example, taking 11.0 pounds and 0.45 surface ratio as a basis, should one designer estimate the E.H.P. required at 5,500 and another at 6,500, the difference in speed of rotation would be 470 and 430 revolutions respectively.

This is a subject upon which much more might be said. The accurate determination of horsepower required to propel a given ship is of the greatest importance, affecting as it does the entire vessel. More especially does this apply to turbine work, where the area of the propeller is such an essential feature in the design. Recourse should be had, whenever possible, to an experimental model tank. As we have tried to show, the matter does not end here; careful attention to the details of a design should not be relaxed for a single moment.—"The Engineer."

NAVAL ARMAMENTS.

Translated by CAPTAIN S. G. SHARTLE, C. A. C.

On fleet plans and shipbuilding activity the "Taschenbuch der Kriegsflotten," 1908, page 381, has the following to say:

The English navy has inaugurated a new era in warship building. By the creation of their so-called *Dreadnought*, they have forced other sea powers to provide similar ships of at least 18,000 tons, with correspondingly powerful guns. European, as well as other nations, have for some time shown by their naval armaments their appreciation of the political

and economical importance of sea power. We therefore see these new giants of the sea building everywhere. France, Russia, Germany, the United States, Japan and even the smaller countries, as those of South America, are constructing them.

Differences of opinion exist only in respect to armament, whether they shall carry, in addition to the light anti-torpedo-boat guns, only the heaviest caliber, or also medium, or two heavier calibers.

Since against such a modern Colossus the former battleships (or armored cruisers)—taken ship against ship—have no prospect of success, a return to smaller battleships is not to be expected.

Therefore the value of the old battleships and also those of today's fleet is declining—corresponding to the increase in the number of *Dreadnoughts*—and the strength of a fleet will be measured in a short time chiefly by the number of the new battleships it includes.

There results a new starting point for the naval equipment of all sea powers.

Those that decidedly surpass in the building of modern ships will be in position to overtake the tardy navies of previous greater strength.

Thus the course of development that the armored ship and its armament has taken, thanks to the changes made by England, offers to the younger and smaller navy proportionately more favorable opportunities for developing than to the older and larger fleet, which has so much more naval material and yet must completely supply itself with the new.

Germany proposes to adapt her building program to the new conditions, as outlined above. The Navy Program of 1908, it is stated on page 382 of the "Taschenbuch," requires the changing of paragraph 2 of the law of June 14, 1900, regulating the building and rehabilitation of the fleet. This paragraph reads:

"Exclusive of ships damaged, battleships are to be replaced after 25 years, cruisers after 20 years."

The new wording is to be:

"Exclusive of ships damaged, *battleships and cruisers are to be replaced after 20 years.*"

The following parts of the paragraph are not changed:

"The spaces of time run from the year of the granting of the first appropriations for the ship to be replaced to the granting of the first appropriations for the replacing ship."

"For the period from 1908 to 1917 the construction for substitution is regulated according to program B."

The above change necessitates a rearrangement of this program. The old and new are here given together (see page 383):

DISTRIBUTION OF THE SUBSTITUTION CONSTRUCTION FOR 1908 TO 1917, INCLUSIVE.

Program B (old).				Program B (new).			
Year of substitution.	Battleships.	Large cruisers.	Small cruisers.	Year of substitution.	Battleships.	Large cruisers.	Small cruisers.
1908	2	...	2	1908	3	...	2
1909	2	...	2	1909	3	...	2
1910	1	1	2	1910	3	...	2
1911	1	1	2	1911	2	...	2
1912	1	1	2	1912	1	1	2
1913	1	1	2	1913	1	1	2
1914	1	1	2	1914	1	1	2
1915	1	1	2	1915	1	1	2
1916	1	1	2	1916	1	1	2
1917	2	...	1	1917	1	1	1
Total.....	13	7	19	Total.....	17	6	19

Other paragraphs of the law of June 14, 1900, which are not changed, will stand as follows:

I. SHIPS.

There shall be

1. *The battle fleet:*—2 fleet flagships, 4 squadrons, each of 8 battleships, 8 large and 24 small cruisers as scout ships.

2. *Foreign fleet:*—8 large cruisers and 10 small cruisers.

3. *Reserve material:*—4 battleships, 4 large cruisers, 4 small cruisers.

II. LENGTH OF LIFE.

(Quoted above concerning length of active life of battleships and cruisers.)

III. COMMISSIONING.

With respect to the commissioning of the battle fleets, the following principles hold: 1. The 1st and 2d squadrons form the active battle fleet; the 3d and 4th squadrons, the reserve battle fleet. 2. Of the active battle fleet, all are kept in commission; of the reserve battle fleet, one-half of the battleships and one-half of the cruisers are kept in commission. 3. For maneuvering, individual ships of the reserve out of commission are placed in commission temporarily.

According to the foregoing law, the German fleet is to include:

38 battleships.
20 large (armored) cruisers.
38 small (protected) cruisers.
144 torpedo boats.

The above tables refer to building to replace old ships. The following table gives the shipbuilding plan of Germany to 1917, including the increases; the latter are marked v:

Year of substitution.	Battle ships.	Large cruisers.	Small cruisers.	Torpedo-boat division.
1908	3	1 v	2	2
1909	3	1 v	2	2
1910	3	1 v	2	2
1911	2+1 v	1 v	2	2
1912	1	1	2	2
1913	1	1	2	2
1914	1	1	2	2
1915	1	1	2	2
1916	1	1	2	2
1917	1	1	1+1 v	2
Total.....	18	10	20	20

Great stress has heretofore been laid upon the fact that the English 30.5-cm., Mark X, L/45 gun is the most powerful used on shipboard. Its projectile weighs 385 kg., has an initial velocity of 884 m.s. and hence an energy of 15,360 m.t. A new type of this gun, L/50, Mark XI, is now under construction for the *St. Vincent* class of ships (three, of 20,900 tons displacement). This gun fires the same projectile as the Mark X, but with an initial velocity of 902 m.s., and therefore an energy of 16,000 m.t. But they are not satisfied even with this, for under "ships' guns of England" is specified as under construction a 13½-inch (34.3-cm.) gun, L/45, which is intended to impart to a projectile of 567 kg. an initial velocity of 860 m.s., an energy of 21,350 m.t.; that is, more than double the energy attained by the German 28-cm. gun, L/40. Whether this statement is correct appears questionable, for, although this gun is to be produced in England, it is intended for the Brazilian navy.

Even this gun, however, is surpassed by an 18-inch (45.7-cm.) gun, L/28, which imparts to a projectile of 907 kg. an initial velocity of 685 m.s.—that is, an energy of 21,738 m.t. This gun is made by the North American firm, the Bethlehem Steel Company, and has the short life of all long guns of large caliber. Certainly this firm has greatly exceeded the first caliber given (35.56 cm.).

While the English and North American factories seek to increase the ballistic performances of their guns by increasing the velocity or the weight of the projectile, the Krupp factory holds fast to the ballistic performances attained by it as long ago as 1901. These results placed them at the head, and were first surpassed by the two guns mentioned above. The Krupp firm set great store on the fact of attaining the same ballistic performance with a gun considerably smaller in weight, and this has been accomplished in an absolutely surprising manner by the new guns, model 1906. The heavy 30.5-cm. gun, L/50, designed for coast artillery, with a weight of 63,400 kg., performs work of 17,500 m.t.; the same work is accomblished by a Krupp gun of only 48,000 kg. The

heaviest gun of the German navy is the old 28-cm. gun, L/40, which weighs 44,000 kg. It possesses an energy of 6,740 m.t. Although the weight of the new gun is not 10 per cent. greater, the work performed by the new Krupp gun is almost 160 per cent. greater. We now get 365 m.kg. of energy for one kg. of weight; this is the highest efficiency that has been reached in artillery, and it is surpassed only by the guns 40 and 45 calibers long of the Krupp firm. The long Krupp rapid-fire guns, model 1906, of all calibers, show the same high efficiency.

It is worth mentioning that the firm of Vickers & Sons has decreased the initial velocity of their 15.2-cm. gun, L/50, from 1,030 to 972 m.s. So the highest initial velocity now is that of the projectile of the 20.3-cm. gun L, 51.6, of the same firm, namely, 1,006 m.s. (not 1,060, as given in the "Taschenbuch" for 1907, through a typographical error). Next stands the Krupp 10-cm. gun, L/50, with 1,000 m.s. initial velocity.

One may wonder whether the movement which England and North America have undertaken with respect to caliber will find imitation in other States. It may be remarked incidentally that the caliber of 18 inches (45.7 cm.) is the largest that has ever been undertaken in artillery, if one excepts the legendary guns, which are supposed to have been used soon after the invention of gunpowder, for example at the taking of Constantinople.—"Artilleristische Monatshefte," March, 1908.

COMMENTS ON ABOVE NOTES BY TRANSLATOR.

It is thought that a more complete comparison of high-power guns than the above may be of interest. To this end the table on page 750 has been prepared.

It will be seen by the table that so far as power, measured in muzzle energy, is concerned, these guns rank as follows:

U. S. Coast	16-inch.	U. S. Coast	14-inch.
Bethlehem	18-inch.	English	12-inch.
English	13.5-inch.	Sweden, Bofors	12-inch.
Krupp	12-inch.	England, Armstrong	12-inch.
Schneider-Canet	12-inch.	U. S. Coast	12-inch.
Skoda-Werke	12-inch.	U. S. Navy	12-inch.
Vickers-Maxim	12-inch.		

Great stress has heretofore been laid upon the fact that the English 30.5-cm., Mark X, L/45 gun is the most powerful used on shipboard. Its projectile weighs 385 kg., has an initial velocity of 884 m.s. and hence an energy of 15,360 m.t. A new type of this gun, L/50, Mark XI, is now under construction for the *St. Vincent* class of ships (three, of 20,900 tons displacement). This gun fires the same projectile as the Mark X, but with an initial velocity of 902 m.s., and therefore an energy of 16,000 m.t. But they are not satisfied even with this, for under "ships' guns of England" is specified as under construction a 13½-inch (34.3-cm.) gun, L/45, which is intended to impart to a projectile of 567 kg. an initial velocity of 860 m.s., an energy of 21,350 m.t.; that is, more than double the energy attained by the German 28-cm. gun, L/40. Whether this statement is correct appears questionable, for, although this gun is to be produced in England, it is intended for the Brazilian navy.

Even this gun, however, is surpassed by an 18-inch (45.7-cm.) gun, L/28, which imparts to a projectile of 907 kg. an initial velocity of 685 m.s.—that is, an energy of 21,738 m.t. This gun is made by the North American firm, the Bethlehem Steel Company, and has the short life of all long guns of large caliber. Certainly this firm has greatly exceeded the first caliber given (35.56 cm.).

While the English and North American factories seek to increase the ballistic performances of their guns by increasing the velocity or the weight of the projectile, the Krupp factory holds fast to the ballistic performances attained by it as long ago as 1901. These results placed them at the head, and were first surpassed by the two guns mentioned above. The Krupp firm set great store on the fact of attaining the same ballistic performance with a gun considerably smaller in weight, and this has been accomplished in an absolutely surprising manner by the new guns, model 1906. The heavy 30.5-cm. gun, L/50, designed for coast artillery, with a weight of 63,400 kg., performs work of 17,500 m.t.; the same work is accomblished by a Krupp gun of only 48,000 kg. The

heaviest gun of the German navy is the old 28-cm. gun, L/40, which weighs 44,000 kg. It possesses an energy of 6,740 m.t. Although the weight of the new gun is not 10 per cent. greater, the work performed by the new Krupp gun is almost 160 per cent. greater. We now get 365 m.kg. of energy for one kg. of weight; this is the highest efficiency that has been reached in artillery, and it is surpassed only by the guns 40 and 45 calibers long of the Krupp firm. The long Krupp rapid-fire guns, model 1906, of all calibers, show the same high efficiency.

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Krupp	12-inch.	England, Armstrong	12-inch.
Schneider-Canet	12-inch.	U. S. Coast	12-inch.
Skoda-Werke	12-inch.	U. S. Navy	12-inch.
Vickers-Maxim	12-inch.		

State.	Caliber.	Length.	Initial velocity.	Weight of projectile.	Muzzle energy.	Weight of gun.
	Inches.		*f.s.*	*Pounds.*	*ft. tons.*	*Tons.*
U. S. Coast	12	L-45	2,500	1,046	45,375	56.6
U. S. Navy	12	L-45	2,700	869	43,960	50.0
England, Armstrong	12	L-45	2,900	850	49,540	58.0
England, Navy	12 / M XI	L-50	2,960	847	51,630	66.0
England, Vickers' Sons & Maxim	12	L-46.5	3,000	850	53,016	57.7
Germany (Krupp), Navy Coast	12	L-50	3,250	981	56,450	47.7 / 52.6
French, Schneider Canet	12	L-50	3,120	827	55,646	57.5
Austria, Skoda Werke	12	L-50	3,150 / 2,790	772 / 992	53,794	60.3
Sweden, Bofors	12	L-50	3,075 / 2,715	772 / 981	50,138	49.2
England, Navy	13.5	L-45	2,820	1,247	68,746	86.0
U. S. Coast	14	L-35	2,150	1,650	52,940	49.5
U. S. Coast	16	L-40	2,345*	2,400	91,500	130.0
U. S., Bethlehem Steel Works	18	L-28	2,250	2,000	70,200	60.0

* On test firing.

As stated, the tendency, in the United States particularly, and in England in one case, is to gain power by increasing the caliber; in Germany and other countries the same end is attained by increasing the velocity. This question need not be gone into here, but the figures of the table stare one in the face. The artilleryman wants: first, a great number of hits in a short time, and, second, great striking energy, since the latter is entirely a waste without the former. Rapidity of hits means: first, flat trajectory, that is, high initial velocity; and, second, a light projectile and gun, the latter not being so important. Great energy means either high velocity or great mass. The latter alternately results in slow fire and less danger space. By the former, high velocity, the artilleryman increases his chances for hits and at the same time the energy. Surely, he wants high velocity, if it can be attained. But it is objected that high velocities wear out guns so rapidly that their efficiency is impaired and their objects, hits and energy, defeated. This is a problem for the gun maker and not for the artillery. Other countries have apparently solved it—the

13.5-inch gun of the English still has 2,800 f.s., and the Krupp 12-inch gun is put down at 3,250 f.s. This gun fires a projectile of 981 pounds and gives a muzzle energy of 56,450 foot-tons as against 52,940 foot-tons energy with a projectile of 1,650 pounds of our 14-inch gun. The English 12-inch gun, with a projectile of 847 and initial velocity of 2,960, falls little short of the 14-inch gun in energy, viz: 51,630 foot-tons. It must be admitted that the English and German guns come nearer the artilleryman's ideal of a high-power gun than does the 14-inch gun.

As to the technical efficiency (*verwertung*) of a gun, expressed in the ratio of work to weight, the coast artilleryman is not much concerned. A few tons more or less matter little in the handling on a stable platform. But even here the German gun is superior, the efficiency of the Krupp seacoast gun being 1,071 foot-tons of work to 1 ton of weight, and of the Krupp navy gun 1,183 to 1; while the efficiency of the 14-inch gun is 1,070 to 1. However, as said before, this is a question of more interest to the constructor than to the artilleryman.

As a matter of historical interest it may be well to correct the statement that the Bethlehem 18-inch gun is the "largest ever made, if the legendary guns of the time of the taking of Constantinople be excepted." As late as the 80's, the United States army and navy each had a few 20-inch smooth-bore guns, which were listed as available for service.

RECENT FRENCH TORPEDO-BOAT DESTROYERS.

The French navy will this year be augmented by a flotilla of twenty-one torpedo-boat destroyers. The first batch of these has recently been completed, and on their trials the vessels attained a mean maximum speed of 29½ knots, which may be considered as satisfactory, having regard to their tonnage. The accompanying Table I gives, under the heading *Sabretache*, the principal data concerning these boats, while under the heading *Mousquet* are given those of a type which

immediately preceded the *Sabretache* class, and under the heading *Chasseur* details of some later vessels which are at the present time under construction. For the purpose of comparison we also give in Table II some corresponding particulars regarding the British coastal destroyers, and the *Tartar*, as well as those concerning the German destroyer *G 137*. A comparison of the figures in these two tables is instructive, and emphasizes the contentions brought forward in the following criticisms, which are those of a Frenchman well versed in matters concerning not only the Navy of his own country, but also those of other powers. He contends that the vessels of the French navy which are called torpedo-boat destroyers, or to use the French term *contre-torpilleurs*, are destroyers in name only. They are, in fact, very similar to the British "coastal destroyers," which are now-a-days called first-class torpedo boats, but are much inferior both in

Table I.—DETAILS OF RECENT FRENCH DESTROYERS.

Nationality	France.	France	France.
Class type	*Mousquet.*	*Sabretache.**	*Chasseur.*
Displacement	302.76 tons	328.70 tons	447 tons
Length at the water load line	184 ft. 8 in.	190 ft. 4 in.	210 ft. 8 in.
Breadth at water line	19 ft. 8 in.	20 ft. 7 in.	22 ft.
Draught (astern)	9 ft. 4 in.	9 ft. 9 in.	7 ft. 9 in.
Boilers	2 Normand or Du Temple	2 Normand	4 Normand
Engines	2 3-expansion; 3 cylinders	2 3-expansion; 3 cylinders	1 3-expansion 3 cylinders at center; 2 turbines at wings
Propellers	2	2	3
I.H.P. (expected)	6,300 to 7,200 on trials	6,800	7,200
Speed on trials (max)	28 (mean) 30.75	29.5, 30.7	28 knots
Bunker capacity	27 tons	30 tons	.....
Radius of action at 10 knots (estimated)	2,300 miles	2,300 miles	2,300 miles
Radius of action at full speed	217 miles	217 miles	.....
Armament	1 of 65 mm. 4 of 47 mm.	1 of 65 mm. 4 of 47 mm.	6 of 65 mm.
Torpedo tubes	2 of 450 mm. (18-in.)	2 of 450 mm. (18-in.)	3 of 450 mm. (18-in.)
Crew	4 officers, 56 men	4 officers, 58 men	.....

* It is worthy of note that this latest type has as protection a deck plating of nickel-steel above the boiler and engine room—this nickel plating is 20 mm. thick (¾-in.), and that the *Chasseur* has similar protection on deck—20 mm. nickel-steel.

These last destroyers are six in number. Three of them will be fitted with a central triple-expansion, three cylinders, main engine and side twin-screw turbine engines. The turbines are of different types—one is of Parsons type, one of the Rateau type and one of the Breguet type. The other destroyers will be fitted only with two main engines of the usual type. It is said that liquid fuel will be used on certain of them (*Chasseur*, *Voltigeur*, *Tirailleur* fitted with turbines).

Mortier, *Fleuret*, *Coutelas*, *Stylet*, *Troublon*, *Cognee*, *Carquois*, *Trident*, *Pierrier*, *Hache*, *Massue*, *Glaive*, *Poignard* (which have been built in the dock yards), *Bramlebas*, *Etendart*, *Fannion*, *Sape*, *Fanfare*, *Gabion*, *Cognee*, *Oriflamme* belong to the *Sabretache* type. Those of the *Spahi* type include *Voltigeur*, *Hussard*, *Tirailleur*, *Carbinier*, *Chasseur*.

Table II.—DETAILS OF RECENT BRITISH AND GERMAN DESTROYERS.

Nationality	England.	England.	Germany.
Class type	Coastal destroyer	*Tartar*	*G 137*
Displacement	215 tons	900 tons	572 tons
Length	166 ft. 6 in.	272 ft.	244 ft. 9 in.
Breadth	17 ft. 6 in.	26 ft.	23 ft. 9 in.
Draught (astern)	5.8 ft.	9 ft.	8 ft. 9 in.
Boilers	2 Thornycroft	4 Thornycroft	4 Schultz
Engines	3 Parsons turbines	3 Parsons turbines	3 Parsons turbines
Propellers	3	3	3
I.H.P.	3,750	14,500	10,000
Max'm speed on trials	27.52	35,672 (mean of 6 runs)	33.10
Bunker capacity	20 tons	74 tons of oil	.....
Radius of action	1,200 miles at 13 knots	1,700 miles at 13 knots	.....
Armament	2 12-pounders	2 4-in. guns in some of the class	1 of 88 mm., 4 of 52 mm. 2 machine guns
Torpedo tubes	3 18-in.	3 18-in.	3 18-in.
Crew	35	68	.....

* The coastal destroyer is now classed in our Navy as a first-class torpedo boat.

armament and speed to the British and German destroyers which were built at the same time. They could be regarded, continues the critic we are quoting, as torpedo-boat destroyers only in French naval maneuvers. The type of modern destroyer of the great maritime nations against which these vessels would have to pit themselves range up to nearly 900 tons displacement, and it is certain that in an engagement the French craft would be little, if any, better than torpedo boats pure and simple.

From the foregoing it is clear, he points out, that these so-called destroyers would have no opportunity of displaying their qualities as destroyers of torpedo boats. Fighting, as they would have to, did they engage with either the British or German flotillas, they would meet with certain destruction, since they would have to deal with enemies which are individually stronger, and, taken collectively in squadrons, more numerous.

Further than this, it is pointed out that the fact that the French destroyers are so different from their counterparts in other navies arises from the fact that in France there is an entirely wrong conception as to the duty which this type of vessel is called upon to perform. So far from having to contend with boats smaller than themselves, they will be opposed to vessels which are superior in every way, and it is added that the function which they are capable to some extent of performing is that of protecting a squadron of larger vessels

from attack—though when due consideration is made of all the factors, they are really not even powerful enough for that duty.

These strictures, coming as they do from a Frenchman, and one who is acquainted to a very considerable degree with naval matters, are undoubtedly severe, but it cannot be denied that the authority we are quoting makes out a very good case against the latest additions to the French flotilla of destroyers.

The following are among his reasons for his statements:

In the first place, the speed of this type of boat is not sufficient. A mean speed of from 32 to 33 knots on service ought, he considers, to be the minimum. Moreover, the weight and the space set aside for machinery and boilers ought to be increased, so as to avoid having delicate boats, whose speed has to be reduced considerably as soon as a bit of sea is met with. At all events, oil fuel and steam turbines should be employed, as these alone will give for a minimum of weight the speed required and the necessary radius of action. The strength of the decks should also be greater than it is, so that under no circumstances can any boat be injured when making use of her armament. Even under the most favorable circumstances it is necessary, in order that a torpedo may be certain of reaching its mark, that it should be fired as near as possible to that mark, so that the causes of error arising from distance, and from the speed and direction of the vessel aimed at, may be reduced to a minimum. To this end it is necessary that the attacking boat should have its stability, speed and facility of maneuvering most perfectly developed, and it is urged that it is impossible to have all these qualities condensed into a hull of only 328 tons' displacement. At least 600 tons would be required.

With such a displacement a type of destroyer might be developed which could maintain a speed of from 32 to 33 knots on service; could carry an armament of three torpedo tubes, three 100-mm. and six 65-mm. guns. Such a vessel could render all the services required of modern vessels of

this type, that is to say, it would be capable of playing either an offensive or a defensive part, and could protect the coasts or accompany fleets.

Our critic considers that the best policy which the Conseil Supérieur of the French navy could pursue is that carried out by the German authorities, for it could not hope to equal England, against which it would be necessary to bring a fleet capable of taking the offensive, whereas in recent years the French fleet has declared itself to be purely for defensive purposes.

At this point he draws attention to several anomalies which exist in the French navy. Until quite recently the chief destroyers of a group of torpedo craft were painted white, while their attendant vessels were painted black. This he ironically presumes was in order to render it more easy for an enemy to discover the presence of a flotilla which was to attack it! Lately, however, the chief vessel of a group has been painted —to use the French expression—*en toile mouillee*, a kind of khaki. It would have been much more simple, it is urged, to have all the hulls painted black or some other neutral tint.

It is also remarked what little progress has been made in the French marine in the use of oil fuel. It is not as though experiments in this direction in France were only a thing of yesterday. The first trial of liquid fuel in that country was made, our informant believes, on the yacht *Aigle*, which belonged to Napoleon III. The matter was then dropped, taken up again, again dropped, and so on. "It is true," he continues, "that use is made of a combination of liquid and solid fuel, but France has not achieved the splendid results obtained in the British navy, of which the scientific spirit, the desire to excel, and the methods of research are admirable, and are the source of its perfection and of the superior work done by this fleet, which is followed in the line of progress by that of Germany. The latter country, although it has only lately started on the conquest of the world, takes a rank

which is continually becoming more important, and the importance of which must go on increasing."

The *contre-torpilleur Sabretache*, has the following characteristics:

Length on the water line	58 m., say 190 ft. 2½ in.
Beam	6.28 m., say 20 ft. 7 in.
Maximum draught	2.96 m., say 9 ft. 8 in.
Corresponding displacement	328.7 tons.
Indicated H.P.	6,800.
Number of boilers	2 (Normand).
Working pressure	265 lbs.
Grate area	162 sq. ft.
Heating surface	7,500 sq. ft.
Number of engines	Two. Triple-expansion, three cylinders, 19 in., 31 in. and 45 in.
Stroke of pistons	23 in.
Revolutions per minute	310.
Speed to be realized during six hours	27½ knots.
Coal consumption required per sq. meter of grate area in full-power trial	400 kilos, = 37 lbs. per sq. ft.

It is remarked that although this type of destroyer, or, more correctly, seagoing torpedo boat, has a slightly greater displacement than its predecessors, its quarters are less comfortable, which is certainly not an advantage, because it is impossible to expect the same work from a tired crew, which can only with difficulty obtain repose, as from a crew which can sleep in greater comfort. There are other points in the design which are far from being in the direction of progress.

The complement of this type of boat is 4 officers and 58 men. The bunker capacity is 30 tons of coal, which gives, according to our informant, a theoretical radius of action of 2,300 miles at 10 knots. The armament is very weak. There is one 65-mm.—2.56-inch—gun in the bows, and six 47-mm.—1.85-inch—guns on the broadside. There are two torpedo tubes, one amidships and one astern, for 450-mm.—18-inch—torpedoes of the latest type.

The *Sabretache* was built in the yard of MM. de la Brosse et Fouche, of Nantes, who have constructed not only the hull, but also the engines and boilers. She was launched in a prac-

tically complete state, with machinery on board. She began her builders' trials only a few days after she was launched.—"The Marine Review."

MANGANESE-BRONZE.

By C. R. SPARE.

Paper read before the American Society for Testing Materials, at convention, June 23-27, 1908.

It is not the intention of this paper to review the history of the development of the various manganese-bronzes up to the present day, but to outline briefly the vastly extended uses to which these valuable alloys have been applied in recent years by the various industries, and, more especially, to point out the methods of testing employed and the physical results obtained.

The best present-day alloys of manganese-bronze are the results of years of scientific research by several of the largest manufacturing concerns in the world, who maintain chemical and physical testing laboratories in connection with their brass foundries. Of all the metal industries, brass founding is almost the last to pass from the empirical and rule-of-thumb to a scientific matallurgical basis. In fact the majority of the brass and bronze manufacturers today have never used chemical analysis or a testing machine, and, therefore, have only a general idea of the composition and physical properties of their products. This is the result of natural conditions, which years ago did not impose severe requirements on brass and bronze.

Today conditions have changed, or are changing, and engineers have multiplied powers and manipulated pressures beyond all former figures. Machinery is more powerful, more complicated and more severely tested than ever before, and while steel and other industries have made great advances and kept apace of requirements, the same ratio of improvement has not been maintained in the copper and brass metals,

excepting in the case of the high tensile-strength manganese-bronzes.

There are several brands of manganese-bronze on the market, each with its own characteristics, which are determined by the chemical composition and the process of manufacture. While upon analysis some of these bronzes do not reveal striking differences in composition, yet these alloys are susceptible of wide variations in physical properties by slight changes in composition, difference in quality of raw materials and methods of melting and mixing.

Certain impurities affect very injuriously the strength and ductility of manganese-bronze, causing brittleness and crystalline texture. It is not within the compass of this paper to discuss these differences, but they become very evident in the testing machine.

APPLICATIONS.

Manganese-bronze first found its important practical application in the manufacture of propeller wheels for ships. Now, cast iron and cast steel have been almost universally displaced by it, not only for all naval vessels, but the merchant ships of the world have adopted it as standard practice. Marine engines have been increased to such vast powers, especially during the past ten years and since the advent of the steam turbine, that 20,000, then 40,000, and now 70,000 I.H.P. are transmitted through manganese-bronze propeller wheels. In some ships the speed runs upwards of 300 revolutions per minute.

PHYSICAL PROPERTIES.

There is no metal of equal strength and toughness which will produce such sound, smooth and intricate castings, true to the form of the pattern. These qualities permit of the maximum fineness of section of the propeller blade, and at the surface speeds of thousands of feet per minute the surface friction is reduced to a minimum.

This bronze is practically incorrodible in sea and alkali

waters. Likewise dilute acids and acid mine water are withstood very successfully. Long-time tests have been made with the various acids and mine waters. Cast blocks of manganese-bronze immersed in the most acid mine water in Pennsylvania, containing about 300 grains sulphuric acid per gallon, for one year, showed no material corrosion, and a test piece cut from this metal showed no diminution in tensile strength over a duplicate specimen previously tested.

TESTING.

Test pieces cut from cast propellers should show an average ultimate tensile strength of 70,000 pounds per square inch, elastic limit of about 35,000 pounds, or one-half the ultimate; elongation in 2 inches, 25 per cent., and reduction of area, 25 per cent. The reduction of area follows the elongation closely in cast manganese-bronze.

These figures can be varied to suit requirements. A soft manganese-bronze having an ultimate strength of 60,000 pounds per square inch and 40 per cent. or 50 per cent. elongation in 2 inches is the lower limit, while an exceedingly hard manganese-bronze can be made to test over 90,000 pounds per square inch, with even as much as 30 per cent. elongation.

The methods of testing manganese-bronze physically do not differ materially from the standard methods for steel. Test pieces, according to the U. S. Government standard sizes, .505 inch, .798 inch or 1 inch diameter, are machined 2 inches between punch marks, and threaded. In the case of castings there is not a sharply defined elastic limit, and the yield point cannot be determined by the drop of the beam. A multiplying divider (such as devised by J. A. Capp), or a standard extensometer will establish quite closely the point at which an appreciable change in the rate of stretch takes place. In the case of rolled or forged manganese-bronze the yield point is more closely defined and the elastic curve is frequently sharp enough to detect the drop of the beam or halt of the gauge wheel.

In compression, cast manganese-bronze, if properly made, shows an average elastic limit of 35,000 to 40,000 pounds per square inch, and a maximum crushing load of 90,000 to 100,000 pounds per square inch. Rolled or forged manganese-bronze tests 50,000 to 60,000 pounds elastic limit, and as high as 130,000 to 150,000 pounds maximum crushing load.

ROLLING AND FORGING.

Manganese-bronze can be rolled or forged readily at a red heat with the production of an exceedingly tough, dense and close-grained metal. Microscopic examination of cast manganese-bronze after polishing and etching reveals a very homogeneous and uniformly grained metal, but after rolling or forging to a sufficient reduction the structure is reduced to from $\frac{1}{30}$ to $\frac{1}{50}$.

Rolling and forging raise the proportional elastic limit to from 45,000 to 75,000 pounds per square inch, depending upon the finishing temperature and the amount of work done on the metal. Likewise the ductility and toughness are increased without, however, a corresponding increase in ultimate strength.

Forged and rolled rods find a wide application as piston rods, shafts, axles, and for all purposes where a metal of equal strength and toughness to carbon-steel is desired, and which will not rust or corrode in the atmosphere, mine or sea water.

An especially soft and tough metal is made to resist vibratory and sudden stresses and shocks. It is used under very severe conditions in modern naval ordnance. It also is finding application in staybolts for locomotives. This bronze tests 40 to 50 per cent. elongation with about 60 per cent. reduction of area.

Thus only a few of the better known applications of manganese-bronze have been pointed out.

The electrical industry is calling for it for turbo-generator sets which are run at speeds up to 4,000 revolutions per min-

ute, and the blades of the steam turbine are satisfactorily made of extruded manganese-bronze shapes. In this case the erosive action of high-pressure steam is a severe condition under which most metals fail.

APPLICATIONS.

The infant industry of this country, which has grown to such lusty proportions, the automobile, has set a very high standard for materials, which must stand up under its peculiarly hard service conditions. Automobiles with gasoline motors of 60 and even as high as 130 H.P., at speed of 1,000 to 1,500 revolutions per minute and running over all kinds of roadbeads at 60 and 100 miles an hour, have stimulated the improvement of the materials entering into the construction more, perhaps, than any other modern machine. The nickel-chrome and other high-grade steels have produced results never heard of before, and cast iron has been wonderfully improved by the requirements of automobile cylinders. Likewise manganese-bronze, formerly used almost exclusively for warships, has been adapted to produce forgings and castings which will not crystallize or fail under similarly severe conditions.

As stated at the outset, it was not the purpose of this paper to explain the theory or history of the deoxidation of copper by means of manganese, but it has been attempted to outline the growing application of these alloys and to present a general idea of the physical tests obtained by several government and private testing laboratories, with the hope that the engineers of this country will become more interested and better acquainted with these alloys.—"The Metal Industry."

THE LUBRICATION OF MARINE-ENGINE CYLINDERS.

Regarding the lubrication of marine-engine cylinders, my experience is that a vertical engine having a piston valve on the high-pressure cylinder will, and does, run as well without internal lubrication as with it. I have operated engines that had a piston valve on the high-pressure cylinder, or on the

high-pressure and intermediate-pressure cylinders, both with and without internal lubrication, and could see no difference, except that in case of the engine using no internal lubrication the piston rods had to be watched a little more closely to prevent heating.

As to using graphite for the purpose, it is sometimes added to the cylinder oil in the swabbing bucket, but I have never seen it introduced into the steam, although I do not see why it would not prove effective.

There is one thing in modern marine engines which I think could be adapted to stationary engines with considerable advantage, and that is the use of liners in the valve chests and cylinders.

The cylinders are cast of ordinary gray iron and fitted with liners of hard, close-grained cast iron. When the liner will stand no more re-boring it may be removed and a new one substituted.

The liner is flanged at the bottom and secured by bolts. It is free to move at the upper end and is made steamtight at that end by a copper wire calked into an under-cut groove between the liner and cylinder, or by a gland and packing space. Of course, it adds somewhat to the initial cost of the cylinder, but makes the cylinder everlasting, and it is much cheaper to renew than a new cylinder would be, while it gives a good, hard-wearing surface for the piston rings to bear on. Then, it may be removed after one light re-boring and the new liner put in, thus obviating the expense of a new piston. The liner is usually given a bearing at the top, at one or two points on the way down and at the bottom, and the spaces around it may be used as a steam jacket or not, as desired.

Possibly the hard, close-grained wearing surfaces may have something to do with the polish found in opening the cylinders. I have been with engines which had no means of putting oil into the cylinders except through the indicator pipes, which was necessary in the low-pressure cylinder upon starting up.—A. T. ROWE, in "Power and the Engine."

OILING ARRANGEMENTS FOR STEAM TURBINES.

The prime importance of getting a perfect oiling system for any type of moving machinery is, of course, too obvious to need demonstration to any practical engineer, but with the exceedingly high rotating speeds of the types of steam turbines in use in this country, the oiling system is even more important than in the case of reciprocating plant, inasmuch as in the case of reciprocating engines it would be most unlikely that more than one bearing at a time would get warm, whether hand or automatic lubrication were adopted. A trouble of this sort, says the "Electrical Review," would probably involve the shutting down of that particular engine until the bearing had been examined, cleaned up, and put into working order again, but otherwise there is no probability of further damage.

The conditions attending the oiling arrangements of the turbine, however, are considerably different. The efficiency of the steam turbine depends very largely on the exceedingly small clearance between the rotor and the casing of the turbine, and should there be any displacement of the rotor with respect to the casing, the turbine blading would be entirely stripped, involving the destruction of the turbine as an efficient factor in the station for the few weeks necessary for reblading. Such a displacement could be very easily caused by the melting out of the white metal in the bearing at either end of the turbine shaft; the extreme importance, therefore, of watching this point cannot be overestimated.

There are two usual methods of lubrication favored by the various turbine builders in this country and America, named respectively the central system and the single-unit system. In the case of the central system a large tank is placed at the highest point of the building, which tank is kept full of oil. Pipework is arranged from the tank to the engine room, through which the oil flows by gravitation to the bearings of the turbines. By means of a parallel system any number of turbines can be supplied from the same oil tank up to its

limit in capacity. In leaving the bearings of the turbines the oil flows into a filter, and after passing through this it enters a series of two or three settling tanks, flowing from one to the other over weirs. From the last tank it is drawn off by means of a pump, and forced up to the supply tank again; and usually on its way up it is passed through cooling tanks through which water is continually circulating, thereby reducing the temperature of the oil almost to the atmospheric temperature.

The chief objection to this system is the danger of a total shut down of the oil plant owing to the giving way of a joint or pipe between the supply tank and the turbine bearings. Moreover, the first cost is particularly heavy, owing the fact that separate steam or electrically-driven pumps are required to keep up the circulation of the oil, and the tanks and pipework have to be of considerable capacity in order to meet all reasonable demands of probable extensions. The quantity of oil required at the first is also very great, sometimes running into thousands of gallons; and should anything occur to spoil the oil, either by an admixture of water or undue heating, the whole batch of oil is spoilt. As a matter of fact, however, the plant appears to work satisfactorily in places where it has been adopted, although the risk taken is probably too great to compensate for any advantage it may have.

The alternative system, in which each turbine has its own oil supply and pump, is probably the ideal one, as it reduces the risk of a total shut-down of the plant to a minimum. The bearings of each turbine are in this system water-jacketed, the water being supplied by one pump operating on any number of turbines required. Moreover, it has the advantage of low first cost, it confines the risk of spoiling oil to each particular turbine; and the system is under the more direct supervision of the engineer in the engine room. Most of the first-class turbine builders at present adopt this method. It is a logical expansion of the principle which has gradually found its way to the front with central-station design for the supply of power of any magnitude—that of the complete unit

system, whereby each piece of plant, from the boiler to the switchboard, is, so far as possible, self-contained, and works independently of other similar sets in the station.—"The Steamship."

PROGRESS OF WARSHIPS AND MACHINERY UNDER CONSTRUCTION IN ENGLAND.

As a leading London daily newspaper in one of its issues during the past month gave its readers what may justly be cited as a sensational description of a monster battleship to be laid down in Portsmouth Dockyard, on the slip to be shortly vacated by the battleship *St. Vincent*, we deem it advisable to assure our readers that no such naval construction has been contemplated by our Admiralty authorities; the detailed description of the imaginary "World's Largest Battleship" being nothing but pure supposition on the part of the writer; seeing that it is stated that she is to be some 3,000 tons heavier than the *Dreadnought*, and is moreover to be propelled by "gas" engines, a type that is so far only in the experimental stage.

That a new battleship will be laid down in Portsmouth Dockyard as soon as the *St. Vincent* takes the water is already known; but no particulars as to her dimensions, engine power, armament or speed are yet known, with this exception, that the vessel is to be an improvement on the *St. Vincent*.

With the foregoing intimation, and giving as heretofore priority of place in our record to the actual work completed and in progress in the Royal Dockyards, we first note that, since the date of the issue of "The Engineer" of January 10th last, such satisfactory progress has been made with the battleship *Bellerophon*, built at Portsmouth, that she gives promise of being ready for commission, complete in all respects—although no "overtime" has been worked upon her—within the two years allowed for her construction.

The unarmored cruiser *Boadicea* being built at Pembroke

Dockyard, and noticed in our record of January 10th last as progressing very satisfactorily in her construction, was successfully launched on the 14th of last May. This vessel is an improved and enlarged scout, being 385 feet long, with a displacement of 3,300 tons, and having at her load-water draught a bunker capacity of 350 tons. She will carry 100 tons of oil fuel. Her turbines, of 18,000 indicated horsepower, drive four propeller shafts, each fitted with a three-bladed screw propeller, cast in one piece with the boss. At full power her turbines are expected to make 500 revolutions a minute, and to propel the ship at 25 knots. Steam is supplied by twelve Yarrow boilers, worked at a pressure of 235 pounds per square inch, reduced at the engines to 170 pounds per square inch, the shipment of which on board was completed in three days—a record piece of work, reflecting great credit on all concerned, as the ship had first to be removed from her moorings to get her into a suitable position under the sheer legs. The *Boadicea* is the first ship built at Pembroke to be fitted with turbine machinery.

The keel plate of a new cruiser of the *Boadicea* class was laid down in a formal way at Pembroke Dockyard on the 15th of June, the laying down consisting in the plates being drawn into position on the blocks by an electric motor, and the first rivet driven into them by a pneumatic arrangement improvised for the purpose. Originally intended as a replica of her sister, the *Boadicea*, she is to have an additional displacement of 50 tons, but is to be of the same draught. The new ship will be 385 feet long between perpendiculars, have an extreme breadth of 42 feet, a displacement of 3,350 tons, and a mean water draught of 13 feet 6 inches, at which her turbine engines of 19,000 horsepower are expected to give her a maximum speed of 25 knots an hour. Her coal and oil-fuel stowage will be 450 tons.

The testing of the condensers, steam pipes and other small details of the armored cruiser *Defence*, built at Pembroke Dockyard, having been dealt with early in the past half year, the preliminary basin trials of her propelling engines have

been made, and were highly satisfactory, both engines having first worked together at a quarter speed, and then separately at half speed.

The battleship *St. Vincent*, the keel plates of which were laid in the last week of December, 1907, at Portsmouth, will at the date of her launch, on the 10th of September next, have been under construction but eight months and a-half. She is of the improved *Dreadnought* type, but has a displacement of 19,250 tons, or 1,350 tons more than the type ship. She is 500 feet long, with a beam of 54 feet and a mean water draught of only 27 feet. Her propelling engines, of 24,500 indicated horsepower, which are to be turbines, are designed to give her a speed of 21 knots under natural draught, steam being supplied by boilers of the Babcock & Wilcox type.

The propelling machinery of the *St. Vincent* is being made by Scott's Shipbuilding and Engineering Company, of Greenock, and rapid progress has been made with it during the past six months. The boring-out operations in connection with the propeller shafting is also proceeding apace at the ship. The work on the steam generators is also in a very forward state. In accordance with the latest Admiralty practice, these are adapted for burning oil fuel in conjunction with coal.

With the *St. Vincent* afloat, and completed, there will then be four ships in active service in our Navy that embody in their design and construction the *Dreadnought* principle; not taking into account several others of the same type being built or not yet completed for sea. The *St. Vincent's* building progress has been extremely satisfactory, as no "overtime" has been worked on her.

The progress of the battleship *Collingwood* at Devonport Dockyard, which was commenced on February 3, a little over a month later than that on the *St. Vincent*, has been very satisfactory. Up to February 8, 700 tons of material had been worked into her hull; by April 25, 2,500 tons; by May 7, 2,900 tons; and in the four working months after the ship was laid down about 3,300 tons. Such work reflects great credit on all employed on her, as the building "staff"

was smaller than usual on a similar sized vessel, and the "slip" not provided with such modern appliances as to enable work to be carried on more expeditiously and economically when on it than when afloat. The ceremony of laying the *Collingwood's* keel plate, which was in two portions, consisted in first riveting the two parts together and then transporting them to their proper position on the blocks. The ship is the sixth battleship of the *Dreadnought* type, and represents the latest ideas in big warship design.

The keel plate of a new armored cruiser of the *Boadicea* class, to be named the *Caractacus*, was formally laid on the building slip at Pembroke Dockyard on the 5th of June. She will be 385 feet long, 42 feet beam, and have a displacement of 3,350 tons, her mean water draught when loaded being 13 feet 6 inches. She will be propelled by turbine engines of 19,000 indicated horsepower to give her a speed of 25 knots, and will be armed like the *Boadicea*.

The ships in the "completing" stage in the Royal Dockyards are the battleship *Bellerophon*, at Portsmouth; the armored and unarmored cruisers *Defence* and *Boadicea*, at Pembroke; and the cruiser *Shannon*, at Chatham; the last vessel will shortly be ready for commission.

Turning, now, to the progress made in the past six months in warship and machinery construction in the private shipyards and engine works engaged in these industries in England, we find that during that interval, owing to the recent strike of shipyard workers in the Tyne district, a steady progress of warship construction, at the works of Sir W. G. Armstrong, Whitworth & Co., at Elswick, has not been maintained. Considerable progress has, however, been made in the work on the armored cruiser *Invincible*. The main armament of the ship is well advanced; the propelling and auxiliary machinery are practically complete, and the usual tests at the moorings have been carried out. The machinery trials are expected to take place shortly. The work on the battleship *Superb* is approaching completion, the fitting of the propelling machinery is in an advanced state, but its further

progress is retarded by the engineers' strike. The Brazilian battleship *Minas Geras* is to be launched in September, the work on board to that end rapidly approaching completion, the boring operations at the stern being finished, and the propeller shafts and propellers fitted in place. The progress of the fast scouts, *Bahia* and *Rio Grande do Sol*, for the Brazilian navy, has been somewhat delayed, but the framing and plating of the hulls are being pressed forward, so that a considerable advance may be looked for during the next six months. Since our last report on work at the Elswick shipyard, the Argentine river gunboat *Parana* has been successfully launched, and the work on board is well forward, machinery, armament and general fittings being well in hand. The sister ship *Rosario*, still under construction, is well advanced, and her launch may be looked for early this month.

Owing the strike of shipyard workers, there is very little to report in the way of warship work effected at Messrs. Hawthorne, Leslie & Co.'s establishments at St. Peter's, or Hebburn, during the past six months. The machinery, &c., of the battleship *Agamemnon*, constructed at St. Peter's, having been previously fitted on board and the sea and other trials successfully carried through, the vessel has been completed and handed over to the Admiralty, and is now commissioned. The strike prevailing in both the engine works and shipyards has delayed the completion of the 33-knot torpedo-boat destroyer *Ghurka*—whose successful trials were recorded in our last report—and she is still in the hands of her builders ready to be opened out for examination before Admiralty acceptance.

The *Afridi*, a similar vessel to the *Ghurka*, but built at Elswick shipyard, fitted with turbine engines by the Parsons Company, and boilers made at St. Peter's—a duplicate of those in the *Ghurka*—has also been laid up owing to the strike. The work on the 33-knot destroyer *Zulu*, an improved *Ghurka*, has just been started at the Hebburn shipyard. The material for the turbine machinery and boilers of the battleship *Collingwood* is all in the workshops at St. Peter's, but owing to the labor trouble no machine work has yet been

done on it. Of the first-class torpedo boats *Nos. 21* and *22*—built—and *Nos. 33* and *34*—building—the first mentioned has been tried at sea, but progress has been stopped owing to labor troubles; while the second vessel, *No. 22*, which has all her machinery on board, is similarly affected. To the same cause no progress has been made in the building of the boats *Nos. 33* and *34* at the Hebburn shipyard since our last report.

Palmer's Shipbuilding Company at Jarrow during the past six months has been engaged on the following work for the Admiralty: The completion of the "trials" of the battleship *Lord Nelson*, which were very successful, the ship having since remained alongside the works for final completion. The construction, still in hand, of the 33-knot torpedo-boat destroyer *Viking;* the successful launching and preparation for her trials of first-class torpedo boat *No. 24;* the similar class boats *Nos. 35* and *36* are now on the stocks. It is to be noted that, owing to the strikes of engineers and shipyard workers, work on the above-mentioned vessels has, as a consequence, been much delayed, and it may be truly stated that but for this the battleship *Lord Nelson* would have been handed over to the Admiralty.

Passing from the Tyne to the Thames, there being no warship work on the *Humber*, we find that such work has also practically left the Thames, Messrs. Yarrow & Co., formerly of Poplar, having migrated to Scotstoun, Glasglow, and Messrs. Jno. I. Thornycroft & Co. from Chiswick to Southampton. Following the usual routine of our notices of warship work completed or in progress, we next record that effected in the past half year by the Southampton firm. Having stated in our last report the actual results attained by the ocean-going destroyer *Tartar*, built at the works at Woolston, Southampton, on her preliminary and official trials, we now record that she was completed and handed over to the British Admiralty last April.

The *Amazon*, another vessel of the *Tartar* type, but with more power and greater displacement, now in hand, will be

launched with boilers and machinery on board in the course of a few weeks, her internal work being in an advanced stage. The *Nubian*, a similar vessel to the *Amazon*, is on the stocks and in a forward condition. Of the torpedo boats being built or completed at Woolston, *No. 19* has completed her official trials very satisfactorily—the contract speed being exceeded—and was handed over early in June. *No. 20* has completed her preliminary trials, and is now ready for her official runs; her completion for sea and delivery will be effected within the next few weeks. Two other torpedo boats in hand for the British Admiralty, *Nos. 31* and *32*, are in an advanced state, and will take the water in the course of the next two months. All the vessels above mentioned are adapted to burn oil fuel, and their turbine machinery has been, or is being, constructed at Messrs. Thornycroft & Co.'s Southampton Works, where in future it is intended to construct all boilers, the shops and plant having been removed from Chiswick for this purpose.

Cammell Laird & Co., Limited, of the Shipbuilding and Engineering Works, Birkenhead, have, since our last record of their work, completed and handed over to the Admiralty the sea-going destroyer *Cossack*. The similar but faster vessel *Swift* is now nearly ready for her trials.

At the shipbuilding works of Vickers, Sons and Maxim, Limited, at Barrow, the keel plate of the battleship *Vanguard*, an improved *Dreadnought*, was laid on the 2d of April. The tender for the building of a similar vessel, the *Rodney*, by the firm, has also been accepted by the Admiralty, and the order to proceed with its construction has been given. This vessel will have a displacement of 20,000 tons, and will be armed with ten 12-inch guns and several others of smaller caliber. She will be the fourth battleship built at Barrow for the British navy.

J. Samuel White & Co., Limited, at East Cowes, have delivered during the past six months the torpedo-boat destroyer *Mohawk* and torpedo boats *Nos. 13*, *14* and *15* to the Admiralty; *No. 16* has finished her trials and is now being

completed for sea; and the destroyer *Saracen*, built by the firm has been launched.

Yarrow & Co., Limited, since their removal from Poplar to Scotstoun, Glasgow, inform us that since the end of last year the second and third destroyers, *Nafkratoussa* and *Lonki*, built for the Greek Government, have left their works at Poplar, and have safely arrived at Athens. The fourth vessel, the destroyer *Sfondoni*, has also been handed over to the Greek authorities, and is expected to leave the Thames shortly. The "Yarrow" boiler constructed for the Spanish gunboat *Marques de Molins* has been forwarded to Lisbon; the ten sea-going torpedo-boat destroyers for the Brazilian Government are still under construction, four of them being well advanced, and one, it is hoped, will be launched in a couple of weeks. The two motor boats constructed for the Austro-Hungarian Government have just run their trials successfully. They are 60-feet long by 9-feet beam, and have a contract speed of 22 knots.—"The Engineer," July 10, 1908.

TORPEDO EXPERIMENTS ON THE MONITOR *FLORIDA*.

The monitor *Florida* was subjected to torpedo fire on the morning of June 13. She was anchored in about fifteen feet of water off Sewell's Point in Hampton Roads. A torpedo loaded with 200 pounds of guncotton was fired from a barge 400 feet distant. The place at which it was aimed was below the armor belt amidships on the port side, and the part of the vessel around it was protected by spars, etc., so that, had the torpedo gone wide of its mark, by any mischance, other parts of the monitor would have been protected. On the interior the vessel was protected with a watertight bulkhead, especially constructed, and a test of which was one of the objects of the experiment.

As the torpedo struck the monitor reeled to starboard, an immense column of water shot up to probably 150 feet in the air, the vessel then settled over to the port side.

An examination showed that a hole "large enough to admit an automobile" had been made in the side below the armor belt, the coal bunker had been blown to pieces, allowing the water to enter and fill the first watertight compartment; two other watertight compartments were damaged enough to admit water, though this damage was not very serious; there was a slight displacement of the armor, but otherwise it was uninjured; the spars and other protection for the vessel in the vicinity of the explosion were wrecked.

The machinery of the vessel was not injured, however. The pumps were immediately set to work and kept the water from entering beyond a certain depth. The *Florida* was taken immediately to drydock at the Norfolk Navy Yard and careful examination made. The results are not divulged by the Navy Department. It is said that the experiment was a success in every way and that valuable information was gained.

The only similar experiments of which there is record were made upon the British ship *Belle Isle*, in February, 1902.

—Abstracted from "New York Times."

TORSION METERS AS APPLIED TO THE MEASUREMENTS OF THE HORSEPOWER OF MARINE STEAM TURBINES.*

By J. Hamilton Gibson.

When a revolving shaft transmits power it always twists slightly throughout its length. In other words, the end at which the power is applied moves slightly in advance of the end where the work is done, the amount of twist varying directly as its length, directly as the moment of the load applied, inversely as the rigidity of the material, and inversely as the fourth power of its diameter, the formula reading—

$$\theta \times \frac{10.2\ TL}{CD^4},$$

where θ is the angular displacement in radius, T = twisting moment in inch pounds, L = length of shaft in inches, C =

* Paper read before the Northeast Coast Institution of Engineers and Shipbuilders, 24th January, 1908.

the modulus of rigidity, and D = diameter of shaft in inches. The law holds good absolutely for all shafts which are not stressed beyond the elastic limit. As shafts are usually designed with a large factor of safety, it follows that the amount of twist, or the "torque," as we prefer to call it, is very small. In propeller shafting, for instance, the torque is rarely more than 1 degree for 10 feet of length, so that for a 12-inch shaft the circumferential displacement is only about $\frac{1}{8}$ inch at full power.

Various methods and numerous instruments have been devised to enable an observer to read off the torque of revolving shafting, and such instruments are rightly termed "torsion meters," or, if self-registering, "torsion indicators." Many of these instruments are extremely ingenious, and it is proposed in this paper to briefly examine and describe some of them.

The rapidly-growing adoption of steam turbines for ship propulsion has created a demand for some ready means of ascertaining their horsepower, and as the steam-engine indicator is not suitable for this purpose, we are thrown back on a torsion meter as the only known method by which such information can be obtained. The power of a steam turbine may be estimated approximately by calculating the amount of water passed by the feed pumps, or by measuring the number of heat units that pass through the turbines in a given time; but a coefficient of efficiency must be first determined, and no account is taken of the revolutions in such estimates. As, however, "revolutions" is the very essence of power in dealing with the question of ship propulsion, that would be a very unsatisfactory method of reporting the power from a shipowner's point of view. How often do we hear the claim made that so-and-so's feed heater, for instance, has given a liner an extra one or two revolutions on the same coal consumption as before? Observe there is no mention of power, because it is recognized that, under similar conditions, the maintenance or improvement of the revolutions is the only thing that matters.

Now, it is a well-known fact that a turbine, unlike a reciprocating engine, passes almost as much steam when standing as when revolving at full speed; and it has, therefore, become an almost imperative necessity, in fixing the responsibility as between the boiler and the turbine, to know what power the turbine is transmitting to the propeller under varying conditions. The power thus ascertained is called the "shaft horsepower," in contradistinction to the term "indicated horsepower," which has come to be applied exclusively to the results obtained by "indicating" the mean pressures in the cylinders of a reciprocating engine. In this connection, "brake horsepower" and "shaft horsepower" are, of course, identical.

A small propeller working deeply immersed in smooth water is a fairly-uniform brake, and the turning moment of a steam turbine is also very uniform. Consequently there is little, if any, fluctuation of the torsional stresses in the propeller shafting. If, then, we can ascertain the torque at only one point in each revolution, it may be assumed that, knowing the revolutions, we have all the information required to calculate the work done. It is very different, however, in the case of reciprocating engines. The turning moment is anything but uniform; there are several points of maximum and minimum torque in each revolution; in fact, it is not an unknown experience to find that at one or more points in each revolution the torque is negative—that is, the propeller, acting as a flywheel, overruns the engine, and actually pulls the engine round after it. In all cases of reciprocating engines, therefore, it becomes necessary to read off the torque at several points in the revolution; the more points the better. The mean torque is then taken in making calculations of power. For a clear appreciation of the problem of torque measurement it is expedient to keep the foregoing facts well in mind, and principally to remember that we are dealing with extremely minute angles, for it is no exaggeration to say that an error of a hair's breadth may mean a difference of several hundred horsepower in the result.

Before applying any form of torsion meter to a shaft we must know its "modulus of rigidity"—that is how much it will twist with a given static load applied at the end of a lever of known length. This can only be done satisfactorily in the workshop, preferably on a long, rigid lathe-bed. One end of the shaft is securely fixed, and a twisting moment applied at the other end. To eliminate the effect of friction in the supporting bearing at the free end it is advisable to use two levers, one at either side, and the loads are preferably applied by graduated spring balances. Two pointers independent of the load levers are secured to the shaft, as far apart as practicable, and the difference in the angular movement of these two pointers gives the true twist for that length of shaft. If the pointers are made 57.3 inches long from the shaft axis, their ends will describe 1 inch of arc for 1 degree of twist, and a decimally-divided straight edge will then measure the twist to within $\frac{1}{100}$ degree, which is quite near enough for all practical purposes, and we can proceed to calculate the modulus of rigidity from the formula.

Observe that a propeller shaft is subject to two distinct stresses. Not only is it twisted as between the engine and the propeller, it is also compressed longitudinally by the propeller thrust, the compressive stress being sometimes as much as 20 per cent. of the shear stress at the surface of the shaft, produced by torsion alone. This compression augments the torque by an appreciable amount, which has been actually measured in numerous experiments, and may be taken roughly as 3 per cent. for hollow shafts and 1 per cent. for shafts which are solid. It might be considered sufficient to calibrate only one shaft in a multiple-screw vessel; but it is found that similar shafts, with identical tensile and elongation tests, have different moduli of rigidity, probably due to their varying elastic limits and some slight difference of homogeneity in the material. The only way, therefore, to ensure accuracy is to calibrate each shaft separately and to build up a power diagram for each.

Another point to bear in mind is that a working propeller

shaft is "alive," and this condition must be imitated as far as possible during calibration, by jarring the shaft with repeated blows of a mallet, so as to keep the mass in a state of molecular vibration. Otherwise the phenomenon of mechanical hysteresis, so marked in some static experiments, will obtrude itself and vitiate the results.

Having established the true modulus of rigidity for each shaft, we may proceed to build up our power diagrams based on the formula

$$H=\frac{\theta D^4 N}{CL},$$

where H = shaft horsepower, θ = torque in degrees, D = diameter of shaft in inches, N = number of revolutions per minute, C = constant varying with the modulus of rigidity, and L = length of shafting in inches. In this formula we have all the elements for obtaining the shaft horsepower, and it only remains to ascertain the number of degrees of torque by means of a reliable and accurate torsion meter. Naturally, a mechanical engineer would employ mechanical means for the purpose—in the first instance, at any rate—and we will describe two such means that have been tried with varying success.

Dr. Fottinger's apparatus has been used on several German boats, and consists essentially of two stiff tubes encircling, but free of, the shaft, except at their remote ends, where they are rigidly secured to the shaft. The free ends of the two tubes are brought together and terminate in a pair of disks, the disk revolving with the shaft in parallel planes. Assuming the disks to be 2 feet in diameter, and the two points on the shaft to which the tubes are secured to be 10 feet apart, the edges of the disks will then have about $\frac{1}{4}$ inch movement relative to one another at full power. Means are introduced to multiply this movement by the employment of links and levers, and the torque is recorded by an indicator pencil moving round a fixed paper cylinder concentric with the shaft. When there is no torque, the line drawn by the pencil is a continu-

ous circle in the same plane, and this line represents the zero, or base line, from which the subsequent torque indications are measured. When the shaft transmits power "ahead," the indication for varying torque, as in a reciprocating engine, is a wavy line on one side of the base line. For "astern" power the indication is, of course, on the opposite side. The backlash of the link work is taken up by light springs to steady the pointer or pencil.

Several diagrams taken by a Fottinger meter were shown, which exhibit clearly the fluctuating torque of shafting driven by reciprocating engines, and at full power a negative torque is seen, as previously referred to, at that period of the revolution where the propeller overruns the engine.

Another form of mechanical torsion meter is that of Mr. Collie. Instead of tubes encircling the shaft, which are limited in length by the distance between the couplings and the plummer-block bearings, two light countershafts, parallel to the main shaft and driven from it at their remote ends by sprocket wheels and chain gearing, are carried overhead. Their free ends are screwed into each other, one of them forming the nut and having a limited longitudinal movement, whilst the other has none, merely revolving in a small thrust block. As one end of the main shaft revolves slightly in advance of the other, the countershafts screw themselves into or out of one another, according to the direction of rotation of the main shaft, by an amount depending on the power transmitted. The longitudinal movement is transferred to a pointer and rendered visible on a dial which is graduated on either side of the zero into so many degrees of torque "ahead" and "astern." The backlash of the gear is taken up by springs, as in the Fottinger meter, and it is a simple matter to add a continuous recording apparatus if such be required.

As a variant on the concentric tube and countershaft methods of measuring the twist of a shaft by means of a parallel member not exposed to torque, several inventors have made use of the fact that some main shafts are hollow and fit an inner shaft loosely fitting the bore. One end of the in-

ner shaft is secured to the main shaft, and to the other end is fitted a pointer or spider, the radial arm of which emerges through grooves cut in the face of a coupling between the coupling bolts. The spider shows the same movement as the remote fast-end of the inner shaft, and moves relatively to the coupling at which it emerges. Various devices are adopted to show and record the relative movement which, of course, gives the torque of the main shaft for the length of the inner shaft.

The best-known electrical torsion meter is the Denny-Johnson apparatus. Briefly, it is made up of two revolving armatures, secured to the shaft as far apart as possible. Each armature has a pointed or chisel-shaped magnet which moves over, but does not quite touch, a finely-wound coil. The coils are connected up in series through a Wheatstone-bridge arrangement to a telephone receiver. When no power is being transmitted, the relative positions of the revolving magnets and coils are identical at each revolution, and no sound is heard in the telephone receiver. But when the shaft twists, the armatures get "out of step," as it were, and a clicking sound is heard until the pointer in the recording box is moved by an amount equal to the number of windings in the coils, indicating that one magnet is ahead, or astern, of the other, until silence again ensues, and thus the angle of torque is caught and measured.

Some time ago Mr. Gardner, of Fleetwood, made an electrical torsion meter based upon the varying amount of current permitted to flow through a wire connected up to an ordinary ammeter. Notched disks, or interrupters, are fitted to the shaft at a reasonable distance apart, and the notches are filled with non-conducting material, so spaced that the conductor and non-conductor are the same length, measured round the periphery of the disks. A brush lies lightly against the edge of each armature, the width of the brush tip being exactly equal to the length of a notch. When no torque is being transmitted, one brush is in full contact on one disk, and the other brush is adjusted so as to be just out of contact on the

other disk. Therefore, the circuit is interrupted, no current flows through the system, and the ammeter stands at zero. Immediately the shaft twists, however, the relative positions of the disks and brushes are altered, and current flows through the system until a maximum is reached, when both brushes simultaneously overlap the conductors by half their width. The widths of the brushes and notches are predetermined to register the full-power torque of the shaft.

Recently Messrs. Barr & Stroud, the makers of the artillery range finder which bears their name, have brought out a torsion meter which is apparently based on the same idea as that of the Gardner apparatus, but no details are as yet available.

Clever and ingenious as these mechanical and electrical instruments undoubtedly are, and manufactured with the greatest possible skill, they all leave something to be desired in the matter of accuracy when it comes to the measurement of power. Every link in the chain between the main shaft and the recording apparatus introduces a possible source of error, and, as has been pointed out, an error, even of the proverbial hair's breadth, means a fairly large percentage error in the horsepower result. In mechanical torsion meters the multiplying gear necessarily involves a multiplication of whatever error there might be; whilst there are more insidious causes of error which creep in in an electrical apparatus, such as variations due to battery resistance, temperature effects, slight dragging of the commutator sections and brush tips, metallic dust or damp on the contact surfaces, and so on.

We turn, therefore, to other methods, and proceed to describe those torsion meters which depend on the action of a beam of light. Herr Frahm, of Germany, and Professor Hopkinson, of Cambridge University, have been working for some time on the same lines, and each has evolved an apparatus so similar that probably the same description will suffice for both. Starting with the concentric tubes and parallel disks of Fottinger's mechanical apparatus or their equivalent, the link work for recording purposes is dispensed with, and a

small plane mirror is used, pivoted to the edge of one disk and oscillated by a projection on the other disc. As the relative movement between the disks increases, so the plane of the mirror is altered. A beam of light from a fixed lamp is projected on to the edge of the disks, and at each revolution of the shaft it is caught on the mirror and reflected on to a graduated scale. In a dark chamber, such as a shaft tunnel, the streak of light from the mirror on the rapidly-revolving apparatus is almost continuous, and the graduations are read off with tolerable ease; but the almost inevitable spreading of the light beam adversely affects the accuracy of the reading.

Mention should be made of a torsion meter invented by Amaler, of planimeter fame. A concentric sleeve is fitted on the shaft, and the free end brought close up to a fixed collar. A short scale is engraved on the collar, and a pointer or vernier on the free end of the sleeve, something like the marks on a micrometer caliper gauge. As the shaft twists, the pointer moves along the scale. The problem now is to read the scale as it is flying round with the shaft. Here advantage is taken of the instantaneous duration of an electric spark. Contacts are fitted on the shaft just in advance of the scale, and a spark throws a powerful light on to the polished scale once in each revolution, so that the scale, however fast the shaft is revolving, appears to stand still, and thus the torque in degrees is read off directly.

The Bevis-Gibson flashlight torsion meter has undergone searching tests during the last eighteen months. Starting with the well-known physical facts that the velocity of light is practically infinite, and that light rays travel in absolutely straight lines through air of even density, it was conceived that some simple means of applying these principles to a solution of the problem of shaft torque should be forthcoming. The usual trial and error work with which inventors are so painfully familiar followed, and eventually the flashlight torsion meter was evolved and put into use. By a mental process of elimination, it was decided at the outset that the less "gear" the better. The angles to be measured are so incon-

ceivably minute, and in a rapidly-revolving shaft the time intervals are so inconceivably short, that nothing but an absolute direct reading can give a true result.

The method adopted can be best shown by a diagram. Two blank disks are mounted on the shaft at a convenient distance apart. Each disk is pierced near its periphery by a small radial slot, and these two slots are in the same radial plane when no power is being transmitted, and there is no twist on the shaft. Behind one disk is fixed a bright electric lamp masked, but having a slot cut in the mask directly opposite the slot in the disk. At every revolution of the shaft, therefore, a flash of light is projected along the shaft towards the other disk. Behind the other disk is fitted the torque finder—an instrument fitted with an eye-piece and capable of slight circumferential adjustment. The end of the eye-piece next its disk is masked, except for a slot similar and opposite to the slot in the disk. When the four slots are set in line, a flash of light is seen at the eye-piece every revolution, and if the shaft revolves quickly enough the light will appear to be continuous. This effect is apparent at anything over 100 revolutions per minute. At lower speeds the flash is seen to be intermittent, but this in nowise affects the accuracy and reliability of the result. At each end of the shaft, therefore, we have what is virtually an instantaneous shutter fixed, be it noted, directly to the shaft, and there is no connecting link or gear between the disk, either mechanical or electrical, except the beam of light which flashes once in each revolution clear through the two shutters. Let us suppose now the shaft to be transmitting power. One disk lags behind the other by a definite amount, and although three of the slots are still in line, the fourth slot—namely, that in the lagging disk—effectually blanks the flash and no light is seen at the eye-piece.

This is where the function of the torque finder comes in. To pick up the light again the eye-piece must be moved by an amount equal to the circumferential displacement of the lagging disk. This is accomplished by manipulating the mi-

crometer spindle of the torque finder, on which is a scale and vernier graduated in degrees. While the scale is fixed its vernier moves with the eye-piece, and the graduations are so marked that by the aid of a simple microscope, conveniently hinged, differences of $\frac{1}{100}$ degrees can be readily discerned. For shafts of ordinary size the scale is set 13.6 inches radius from the center of the shaft, so that the degrees are about $\frac{1}{4}$ inch apart. One-hundredth of a degree, therefore, means one-hundredth of $\frac{1}{4}$ inch, or $\frac{1}{400}$ inch. As an ordinary shaft twists 1 degree in 10 feet at full power, it is, therefore, possible to get the shaft horsepower to within 1 per cent. of full power. But as it is frequently possible to fit the disks 40 feet or 50 feet apart, even this accuracy may be improved upon, and powers ascertained to within one-quarter of 1 per cent. of full power. Users of the ordinary steam-engine indicator will readily appreciate what this means, for indicated horsepowers are frequently woefully erratic. When we consider that a steam-engine cylinder is often some thousands of times greater in area than the small indicator piston, we get some faint notion of the effect of multiplication of error. Add to this the friction of the engine piston, the piston rod, the guides, and the connecting-rod joints, and we begin to realize how much more reliable is shaft horsepower than indicated horsepower. For purposes of scientific data, especially in reference to ship propulsion, the latter term will, no doubt, soon become obsolete, and retire into the comparative obscurity of nominal horsepower and other like terms.

The slots in the torsion-meter disks are necessarily of appreciable width, and in moving the torque finder over, the light is visible for some distance along the scale. The light comes into view, attains a maximum amplitude and brightness, and fades away as the eye-piece moves along the scale. If it were possible to gauge the exact point where the light attains the maximum, that is the point that would be used. Failing this, however, use is made of one edge of the slot. The finder is moved always in the same direction in taking readings, and stopped at the exact point where the light is cut off. So del-

icate is the sense of sight that a movement of $\frac{1}{100}$ degree is sufficient to mark the difference between light and darkness. A zero reading is taken when the shaft is revolving idly—if possible, at or near full speed—and this reading forms a base and is subtracted from any subsequent power readings.

Let us see how this works in practice by employing a mechanical lantern slide. First, suppose the shaft to be revolving idly. The finder is moved over until the light is just disappearing, and the vernier is seen to be standing at 0.53 degrees. Now suppose the shaft to be transmitting power. The disks have twisted relatively to one another, and no light is seen until the torque finder is moved the same amount. Having picked up the light the finder is worked gently over until the light is just disappearing again. The reading is now 2.39 degrees. Subtract from this the zero reading of 0.53 degrees and we get the true torque—namely, 1.86 degrees. Now to apply our shaft-horsepower diagram. We will suppose that the revolutions are 475 per minute. The torque is 1.86 degrees, and, finding the intersection of the lines on the diagram, the power is seen to be 3,620.

It is, perhaps, scarcely necessary to point out that the whole operation takes much less time than its description. Indeed, it is possible to produce the shaft horsepower on a trial trip immediately on the termination of each measured-mile run, and to hand a slip to the officer in charge, containing all the information, in plenty of time for him to make any necessary adjustments before coming back on the straight for the next mile.

For reciprocating engines a simple modification of the flash-light torsion meter enables the operator to take several readings—usually twelve—in one revolution of the shaft. The disks are perforated with twelve slots arranged in the form of a spiral—one at each 30 degrees of the circumference. The lamp and torque finder must be moved radially from the shaft, so as to bring them into line with each corresponding pair of slots in the disks. By spotting the readings on a sheet of squared paper and sketching in the curve, we get an actual

twisting-moment diagram, from which the mean torque is readily obtained. The mean torque with the revolutions is then referred to the power diagram, and the shaft horsepower read off as before.

Table I.—FLASHLIGHT TORSION-METER RESULTS.

Engines, No. 1215.

Vessel, H.M.S. *Indispensable.* Date, 31st December, 1907.
Trial, Official Full-Power. At Clyde.

Shaft.	Run.	Steam at H.P. receiver, lbs. per sq. in.	Revolutions per minute.	Reading.	Torque, degrees.	Shaft horsepower.
Starboard, wing.....	IV.	180	710	5.90	1.26	8,300
center			680	6.02	1.20	7,480
Port, center			687	4.64	1.21	7,520
wing.....			707	5.74	1.23	8,270
Mean revolutions.....	...	...	696	Total.....		31,570

The modification of the apparatus is only required in the case of shafting driven by reciprocating engines. In a recent vessel a comparison in this connection was made. A crank-effort diagram was built up, in the usual way, from the indicator diagrams, due allowance being made for the effect of the inertia of the moving parts, and the torsion-meter twisting-moment diagram drawn down to the same scale. The latter curve corresponds closely with the crank-effort diagram, but the variation from the mean is greater in the shaft-torque diagram, due, probably, to the action of the propeller and the torsional oscillations thus set up. It will be noticed that the dotted shaft diagram is consistently below the crank-effort diagram, the mean difference being about 10 per cent. This difference corresponds almost exactly with the result obtained by steaming, and indicating the engines disconnected from the propeller in the wet basin before the underway trials, and forms a striking check and corroboration of the two curves. Cases sometimes occur, especially in modern warships, where

a long length of shafting is not available for torsion-meter purposes, and recourse must be had to a special form of apparatus.

To meet this contingency another modification of the flashlight torsion meter is used, in which the beam of light, instead of flashing axially along the shaft, is made to flash radially through slots in concentric drums, and is caught by a torque finder at a distance from the shaft. The drums are fixed to the shaft only 2 feet or 3 feet apart, and the relative movement due to shaft torque is naturally very much less than that of the discs in the actual form of torsion meter. A masked lamp is fitted inside the smaller drum next the shaft and so close to the drum that when the shutter opens the source of light is exactly at the shutter. The outer drum is made as large in diameter as can be conveniently arranged, the radial distance between the drums, as compared with the distance of the torque finder from the source of light, giving by direct proportion the required multiplication of effect, and enabling the torque, as before, to be read off with extreme accuracy, considering the short length of shaft available. The light in this case is cut off by the three knife edges—one at the lamp, one at the inner drum and one at the outer drum —the eye-piece being fitted with a diaphragm pierced by a minute pin hole in the center. The extreme sensitiveness of the apparatus is almost incredible. The angle of the flashing beam proceeding radially from the shaft can be measured to $\frac{1}{1000}$th of a degree, so that although only 3 feet of shafting may be available, the result is as good as if a 30-foot length had been used with an axial-ray apparatus.

Radial-flash torsion meters are not quite so simple in construction as the axial-flash type; but there are certain obvious advantages besides its applicability to a short length of shafting. For instance, the flash might be led vertically upwards through a tube in a deck immediately overhead, and the readings taken at will in the seclusion that a cabin grants, instead of in the engine room or tunnel. In the practical application of the flashlight torsion meter to various vessels

fitted with steam turbine installations some very interesting results have been forthcoming, which are set out in tabular form in Table II.

Table II.—TURBINE STEAMER.

FLASHLIGHT TORSION METER. ACTUAL READING AND CORRESPONDING HORSEPOWERS TAKEN DURING TRIAL TRIPS UNDER VARYING CONDITIONS OF DISPLACEMENT AND PROPELLERS.

Turbine shaft.	Torque, degrees.	Revolutions per minute.	Shaft horse-power.	Shaft horse-power, total.
Starboard, low-pressure	1.43	482.9	2,775	7,975
Center, high-pressure	1.69	461.2	2,600	
Port, low-pressure	1.37	472.8	2,600	
Starboard, low-pressure	1.32	461.2	2,410	6,940
Center, high-pressure	1.65	426.8	2,330	
Port, low-pressure	1.24	457.3	2,200	
Starboard, low-pressure	1.15	426.4	1,970	5,960
Center, high-pressure	1.52	417.6	2.080	
Port, low-pressure	1.13	418.9	1,910	
Starboard, low-pressure	1.05	418.4	1,765	5.555
Center, high-pressure	1.52	422.3	2,120	
Port, low pressure	1.02	415.5	1,670	
Starboard, low-pressure	0.21	198.6	162	495
Center, high-pressure	0.27	206.3	185	
Port, low-pressure	0.19	183.5	148	
Starboard, low-pressure	0.22	146.7	88	257
Center, high-pressure	0.21	171.4	87	
Port, low-pressure	0.13	144.8	82	
Starboard, low-pressure	0.07	46.3	13	37.2
Center, high-pressure	0.05	86.1	15	
Port, low pressure	0.01	24.4	9.2	

Attention is specially directed to the immense range of the apparatus. Some of the low powers recorded are less than half of 1 per cent. of the full power. If indicated horse-powers of such small amount were required to be taken from a piston engine, the indicator spring would have to be changed for a very weak one to get a reasonably accurate card; but no such change is required in the apparatus we are considering.

Then, again, the distribution of power in a turbine installation can only be approximately estimated. The steam is turned into the high-pressure turbine and left to follow its own devious course through the successive turbines on its way to the condenser. At low powers it is sometimes found that the high-pressure turbine shows the most power, whilst for overloads the lower-pressure turbines have the advantage. The percentage distribution of power is shown by three sets of figures—for a three-shaft turbine installation, including high-pressure and intermediate cruising turbines. Set A shows the estimated or designed distribution, set B the calculated distribution from the pressure-gauge readings at the terminals of each turbine, and set C shows the actual distribution of power over the three shafts as ascertained by flashlight torsion-meter readings.

Referring again to Table II, it will be seen that the starboard low-pressure turbine shows throughout the series more power than the port. Investigation showed that the blade-tip clearances of the two turbines differed slightly, and a further comparison proved that the percentage difference of clearance was just sufficient to account for the differences of shaft horsepower recorded.

In a recent progressive trial of a vessel fitted with triple-expansion engines, flashlight torsion-meter readings were taken at varying speeds. Plotting these results in the manner before described, we notice an almost alarming fluctuation of torque as the power increases, and at one point—namely, where the intermediate crank is at right angles coming up, and the low pressure has just opened full to steam on the downstroke, the high pressure being just past cut-off—the propeller overruns the engine and the torque is negative.

Other observations and comparisons might be made; but enough has been said to indicate the advantages and possibilities of shaft horsepower results, and we must conclude that, whichever type of torsion meter comes into general use on the inexorable principle of "the survival of the fittest," the torsion meter in some form or other has come to stay.

THE INTERNAL-COMBUSTION ENGINE.

By W. G. WINTERBURN, M. I. N. A., Victoria, B. C.

The probability of motor launches being added to the equipment of modern steamships in the near future necessitates the addition of yet another subject to the enclyclopedic knowledge required of the present-day marine engineer. For those who have not given the subject much thought as yet, I propose to give an elementary description of the gasoline engine as used for propelling small vessels.

General principle.—The action of the internal-combustion may be likened to the discharge of a gun: imagine the cartridge to have the shell extended forming a trunk piston; a connecting rod is suspended from the nose of the bullet, its other end attached to a crank outside the barrel; on the crankshaft is fitted a heavy flywheel.

When the detonator is struck it explodes the charge, the reaction of which drives out the bullet, but, being trammelled by attachment to the crank, transmits its energy to the flywheel, which in turn drives the bullet back up the barrel. In the motor, the piston on its return stroke draws in a mixture of gas and air, which is compressed to a point which renders it highly explosive; at this juncture the electric spark, corresponding to the cap in our simile, is flashed and explodes the gas; this gives the piston the impulse which puts kinetic energy into the flywheel for returning the piston, besides doing the work required of the engine.

In the four-cycle motor the charge is admitted to the cylinder and exploded, driving the piston to the end of the stroke —cycle 1. The momentum of the flywheel drives the piston back; during this period exhaust valves open and the spent gases are driven out of the cylinder—cycle 2. The energy still left in the flywheel drags the piston outward another stroke, during which inlet valves open and admit a new charge of explosive mixture—cycle 3. The flywheel again drives the piston back, compressing the gas ready for a new explosion—cycle 4.

In the two-cycle motor the incoming gas is made to sweep out the old and spent gas and an explosion is obtained every revolution; it is not a good mechanical proposition but works well for small power, and as it does away with a number of valves at the attachments for operating them, the mechanism is simplified and first cost reduced.

The operation is this: The gas is admitted and exploded, driving forward the piston; at a point in the stroke the piston uncovers a port in the cylinder wall which permits egress of the exploded gas; another port on the opposite side is uncovered through which rushes new gas which had been confined under pressure, a deflector is cast on to the top of the piston which directs the rush of new gas to the cylinder head, whence it recoils with sufficient rapidity and force as to drive before it the spent gas without commingling with it to any appreciable extent; the return stroke of the piston compresses the new gas, which is then exploded and another impulse given to the piston.

It will be seen that the two-cycle engine requires a lighter flywheel and considerably less mechanism than the four-cycle, but it is not reliable for varying loads and will not stand up to continuous heavy work so long as the latter; also, it is less economical in fuel, as a portion of unspent gas must pass off with the exhaust, whilst a residuum of the latter remains in the cylinder, causing incomplete combustion by adulterating the explosive mixture.

Reversing.—Generally, the engine is intended only to run in one direction, the astern motion being effected either by changing the angle of the propeller blades or by a reversing clutch. There are many forms of reversible propellers on the market, any of which may be applied. The reversing clutch consists of a train of gears, by means of which the direction of rotation of the shaft is reversed, and is identical with the arrangement used in motor cars. In practice, the solid propeller with reversing clutch has been found the more efficient of the two systems, but the noise made by the gearing is objectionable.

The engine itself will reverse if the explosion is timed at the right moment; if, for instance, the revolutions are in the direction of the hands of a clock and the spark is ignited at two minutes past twelve, the piston receives its impetus and the crank revolves in the normal direction; if now on the return stroke the spark ignites at two minutes *before* twelve, the crank will not turn the center, and the piston is forced back, and as long as the spark occurs at this point the engine will run that way. In practice it was found that motors could be reversed by this means when running free, but not with a load on; so a releasing clutch has been devised which disconnects the shaft while the position of the sparker is being altered, it is thrown into gear when the engine starts to go in the direction required.

Cooling.—The heat generated in the cylinder is great and is usually dissipated by means of water circulating round it; in land engines circulation is obtained by gravitation, a cooling tank is placed a few feet above the engine into which the hot water discharges, the cooler water returning to the jacket from the bottom of the tank; for marine work a pump is necessary. Care must be exercised in frosty weather to see that jackets are drained when the engine is not being used. In very cold climates winged radiators cast on the cylinder are preferable to water cooling.

Vaporizer, as its name implies, is an instrument for changing the liquid fuel to the gaseous form; it consists of a brass vessel containing a float and a needle valve; these are so adjusted that air is admitted in the exact proportion to the quantity of gasoline passing through the valve, in order to make the correct mixture for explosion. If an excess of gasoline is supplied the mixture is said to be "too rich," and either the cylinder is flooded with liquid gasoline and the electric terminals wetted, killing the spark, or incomplete combustion takes place and black smoke of offensive odor is emitted from the exhaust, and the working of the engine is weak and erratic, and if too much air enters the gas will not explode at all. An induced draft to the cylinder is created

by the motion of the piston and the gasoline is drawn through fine perforations in the vaporizer, which breaks it into a minute spray when the mixture with air takes place. Once the float-and-needle valves have been correctly adjusted they should not be interfered with except when atmospheric or temperature changes make re-regulation necessary. A butterfly-throttle valve forms part of the device which is operated by a hand lever and the speed of the engine is thereby controlled.

Sparking.—Although theoretically explosion will occur automatically when a given pressure and temperature agree with the right chemical combination, in practice all the precise conditions cannot be relied upon to ensure regular action. An electric spark is utilized to ignite the gas; a plug containing insulated wires is screwed into the cylinder end. The terminals consist of two platinum points with a gap between of about one sixteenth of an inch; the current is conducted through cables to the plug, first passing through the timer. This is a device operated by the engine which opens and closes the circuit; when the circuit is closed the electric fluid jumps the space between the terminals of the plug and a spark is generated in the same manner as the arc light. By altering the position of the timer an early or late spark can be obtained and the explosion timed to take place at the moment most suited to the conditions. Sometimes, instead of the plug, what is termed the "make-and-break" system is adopted; this is an arrangement whereby a reciprocating part of the engine causes a pawl to oscillate within the clearance part of the cylinder; on this is fitted a platinum point which bears against an electrode screwed through the cylinder cover or head; when the movement of the pawl breaks the contact with the electrode the current tries to follow and so creates a spark. The heat and intensity of the spark is a most important desideratum. When two spark plugs are fitted to one cylinder much better combustion results, besides the advantage of increased reliability due to duplication.

Electric current is generated in the first instance in a set

of dry cells, or a battery of the Lalande type. Four cells of about 18 ampères make a good battery for a 3-H.P. motor; these are either placed in a convenient locker and connected in series, or the set can be obtained fitted into a box and encased with hard wax; this is a portable and watertight arrangement, and very suitable for marine work on account of the perfect protection from damp and other destructive agents prevalent on board vessels. The life of dry batteries depends a great deal on weather conditions and changes of climate; they will not last longer than a few months, even when the motor is little used and not consuming current, and as they die practically of senility, it does not avail to carry spare cells, as they also will be found weak when wanted. Being cheap and easily replaced is in their favor when near sources of supply, but for remote places or long voyages a renewable battery is desirable.

The Lalande type consists of jars containing zinc and copper-oxide plates suspended in a solution of caustic soda; on this floats a thin layer of oil to check evaporation. These batteries retain their voltage until exhausted, and when it is necessary to renew the constituents it is an easy matter to remove the wasted metals and pour in fresh solution; the battery is then as good as new. Renewals can be kept indefinitely, as, of course, no chemical action takes place until they are combined.

For continuous working, a magneto or sparking dynamo is often fitted; the engine is started with current from the chemical battery, and when fairly away this is switched off and the magneto brought into operation. The best way of driving this is by friction pulley bearing on the flywheel and held thereto by springs. Belts require constant tightening and gearing is noisy.

The coil.—The current generated by the batteries is of low potential and has to be transformed into high tension in order to produce spark of the required intensity. The current is led by "primary" insulated cable to the coil, which gathers up electro-magnetic energy which is released when the vi-

brating points are separated; the "secondary" cable conveys the current thence to the spark plugs or electrodes within the cylinder. In multiple-cylinder engines special coils are required with terminals for each, or individual coils for each cylinder, the units being interchangeable.

Lubrication.—In all high-speed engines this is a most important factor. For the internal parts care must be taken that only best quality gas-engine oil is used, and the oil must be fed continuously. It passes through sight-feed lubricators and the drops can be regulated with the greatest nicety. An excellent method of feeding is where a reservoir, holding, say, a quart, is fitted in a convenient corner having a pipe connection to the crank case, where always exists a slight pressure; this forces the oil up to the sight-feeds and the amount admitted to the cylinder, etc., is regulated by stop cocks. The wrist pin passing diametrically through the piston is bored hollow and gathers oil from the oil orifice in the cylinder wall as it passes. By this means not only the pin is lubricated, but the surplus oil flows down a channel in the connecting rod to the crank pin; the drippings from this fall into the crank case, which, being hermetically closed, retains it, and the crank splashes it over all working parts. In the two-cycle engine, where the explosive vapor is drawn into the crank case previous to passing into the upper or working end of the cylinder, it becomes impregnated with atomized oil and arrests thereby the smooth running of the piston.

General.—When well cared for, few adjustments are found necessary in well-designed motors and they will run a long time before requiring overhaul. The thrust of the propeller shaft is taken by ball rings on the bedplate; bronze alloys are more suitable for bearings than white metal. The connecting rod in the smaller sizes is of bronze throughout, of H section; the nuts should be very securely locked and pinned; the piston is very deep, having three Ramsbottom rings above and one below the wrist pin. A muffler is necessary to drown the sound of exhaust, some engines exhaust under water, but the back pressure somewhat checks the speed. A governor

fitted within the flange of the flywheel is a useful adjunct, though not always supplied, many makers contending that single-cylinder engines are so carefully balanced as to render governors unnecessary. Multiple engines have the turning moment well distributed; this, however, does not prevent racing, and launches carried on ocean steamers do not always have smooth water to sail in.

In all parts of the world where gasoline is procurable at a reasonable price the internal-combustion engine is displacing steam for small craft. It is not necessary to recapitulate its advantages, and the objection formerly adduced that it is not perfectly dependable is answered by the fact that motor boats have crossed the Atlantic under their own power and without convoy, and motor engines are being fitted into lifeboats, which have provision for self-righting in case of capsize; the engine being enclosed in a watertight compartment and manipulated from the outside by means of rods and handles passing through stuffing boxes in the bulkhead. An engine that will comply with such conditions has surely come to stay.

NOTE ON THE USE OF SUPERHEATED STEAM WITH MARINE ENGINES.*

By Monsieur Felix F. T. Goddard, Member.

Superheated steam was used in marine engines more than half a century ago, after Hirn's noteworthy experiments with the *Logelbach's* engine in Alsace. The French Navy also tried it on some of their earliest protected cruisers.

These early attempts were not, however, followed up, as it was found difficult to construct superheaters capable of maintaining a constant and sufficiently high temperature, and also because of the wear and tear of the hemp packings in use at that period.

The introduction of compound marine engines, more eco-

* Read at the Spring Meetings of the Forty-ninth Session of the Institution of Naval Architects, 9th April, 1908.

nomical than the simple engines that had preceded them, caused the use of superheated steam to be given up for the time being. The same thing occurred with stationary engines, where improvements in valve gear enabled a high ratio of expansion to be employed and the clearances to be reduced to a very small percentage of the cylinder volume.

In the Vosges and Alsace, however, the problem of using superheated steam in stationary engines was revived some fifteen years ago. Several different arrangements were designed by Mr. E. Schwoerer, a former assistant of Hirn's, who used a massive superheater placed behind the fire bridge of the boiler furnace, and this gave such promising results that the study of the question of superheated steam was taken up by a number of manufacturers, chiefly in Germany, Alsace and Switzerland.

It was found that engines fitted with Sulzer Colman valves, which were largely employed in those countries, were very suitable for use with superheated steam. In France, where the Corliss gear was usual in stationary engines, superheating did not make much progress, because it was not suited to the Corliss engine, or, in fact, to any flat slide-valve engine. The exhibits at the Paris Exhibition of 1900 showed this to be the case.

Since then the use of superheated steam with stationary engines has increased largely, and considerable economy has been effected thereby. It is not unusual to find engines of 1,500 I.H.P. to 2,000 I.H.P. using steam at 300 degrees centigrade (572 degrees Fahrenheit) and working at an expenditure of only 4 kgms. (9 pounds) of steam per I.H.P.* per hour.

There is but little information, however, on the subject of the variation in the consumption of steam in relation to its temperature. A few years ago the author made some experiments on a triple-expansion engine with piston valves, the temperature of the superheated steam varying from 0 to 120 degrees centigrade. The results of these trials still present

* I.H.P. referred to in this paper is French "force de cheval" of 75 kgms. = .986 British I.H.P.

some features of interest. In the diagram the abscissæ represent the amount of superheat, *i. e.*, the difference between the actual temperature of the steam (in degrees centigrade) and the temperature corresponding to the pressure when the steam is saturated. The ordinates represent the weight of steam consumed per effective brake horsepower.

TRIPLE-EXPANSION ENGINE.

Diameter of cylinder (H.P.)	0.20 m. ($7\frac{7}{8}$ in.)
(M.P.)	0.33 m. (13 in.)
(L.P.) 2 of	0.37 m. ($14\frac{9}{16}$ in.)
Stroke	0.29 m. ($11\frac{7}{16}$ in.)
Revolutions	440
Power	300 B.H.P.
Cut-off	0.7
Ratio of expansion	9.8
Pressure in main steam pipe (A)	12.8 kg. (182 lbs. pr. sq. in.)
(B)	15.0 kg. (214 lbs. pr. sq. in.)

A exhaust to atmosphere.
B exhaust to condenser.

Now, the curve A in the diagram (exhaust to atmosphere) shows that the consumption per hour of saturated steam (*i.e.*, with no superheat) is 8.85 kg. ($19\frac{1}{2}$ pounds) per B.H.P., whereas it falls to 5.70 kg. ($12\frac{3}{4}$ pounds) at a temperature of 320 degrees centigrade (608 degrees Fahrenheit), equivalent to a superheat of 120 degrees centigrade (216 degrees Fahrenheit).

The saving, therefore, amounts to—

$$\frac{8.85-5.70}{8.85}=35.5 \text{ per cent.}$$

Taking curve B (exhaust to condenser), the consumption per B.H.P. falls from 7.15 kg. (16 pounds) with no superheat to 4.85 kg. ($10\frac{1}{2}$ pounds) with a superheat of 120 degrees centigrade, or a saving of—

$$\frac{7.15-4.85}{7.15}=32 \text{ per cent.}$$

It will be seen, therefore, that superheating may lead to an economy, as compared with saturated steam, of 35 per cent. in engines of this type exhausting to the atmosphere, and 32 per cent. for those exhausting to the condenser.

The amount of reduction in steam consumption depends, of course, upon the design of the engine under consideration; in the present case it amounts to about 1 per cent. for every 4 degrees centigrade (7.2 degrees Fahrenheit) of superheat. This is a figure frequently given, and which the author has been able to verify elsewhere.

Doubts, however, have often been expressed in regard to the efficiency of superheating in actual practice. These arise from the wear of the valve gear of the engines, which causes losses that neutralize part of the economy obtained by using superheated steam.

It is now recognized by all the makers of land engines that, in order to use steam at a temperature of 280 degrees to 320 degrees centigrade (536 degrees to 608 degrees Fahrenheit), the engine friction must be as small as possible, as, for instance, in reciprocating engines with lift valves, and still more so in turbine engines.

Superheated steam is now generally adopted for land en-engines by reason of the economical results obtained in practice, which, in certain special cases, have effected a saving of upwards of 33 per cent.

For marine engines the case is very different. England was the first country to take up the matter; in 1900 Messrs. Wilson & Sons, of Hull, installed superheaters on board the S. S. *Claro*, which appear to have given satisfactory results. Other installations followed with varying measures of success. The British Admiralty investigated the question on the *Britannia* with satisfactory results.

Superheating has also been adopted in the United States on the S. S. *Creole*, fitted with Curtis turbines, and in Germany on the *Ersatz Komet* with Parsons turbines. Nevertheless, superheating in marine engines cannot be said to have gained ground as rapidly as was expected. It may, therefore, be of interest to record some very encouraging results which have been obtained in France within these last few years.

In 1906 the Société de Saint-Nazaire built two identical cargo boats for the Compagnie Générale Transatlantique.

They were the *Garonne*, fitted with ordinary triple-expansion engines and slide valves, and the *Rance* with similar engines, but with the Lentz valve gear. The boilers of the latter vessel are similar to those of the former, except that they are fitted with "Pielock" superheaters.

The leading dimensions of these two ships, their engines and boilers, are as follows:

Length	91.0 m. (298 ft. 6 in.)
Beam	12.20 m. (40 ft. 0 in.)
Molded depth	7.75 m. (25 ft. 5 in.)
Load draught	6.40 m. (21 ft. 0 in.)
Gross tonnage	2,700 tons.

ENGINES.

Diameter of cylinders, (H.P.)	0.584 m. (23 in.)
(M.P.)	0.914 m. (36 in.)
(L.P.)	1.498 m. (59 in.)
Stroke	1.066 m. (42 in.)

The boiler installation of each ship consists of two cylindrical boilers, each having two furnaces and fitted with Howden's forced draft.

	Garonne.	*Rance.*
Total grate area, sq. m	8.40 (90.42 sq. ft.)	8.40 (90.42 sq. ft.)
Heating surface, sq. m	350.08 (3,767 sq. ft.)	277.08 (2,982 sq. ft.)
Superheating surface, sq. m	.....	73.00 (785 sq. ft.)
Total heating surface, sq. m	350.08 (3,767 sq. ft.)	350.08 (3,767 sq. ft.)

The trials of these two vessels were carried out under conditions as similar as possible, so that the comparison of results might be quite fair, and the coal used was the same in both cases. The results of the trials were as follows:

COAL-CONSUMPTION TRIALS.

	Garonne.	*Rance.*
Date of trials	July 6, 1906.	Sept. 13, 1906.
Boiler pressure	12.6 kg. (178 lbs. per sq. in.)	12.54 (177 lbs. per sq. in.)
Steam temperature	192° C. (377.6° F.)	270° C. (518° F.)
Revolutions	72.3	75.37
I.H.P	1,104.	1,304.
Coal consumption per I.H.P.	511 gr. (1.12 lbs.)	408 gr. (0.9 lb.)
Advantage of superheating:		
Increase of power		18.1 per cent.
Reduction of coal consumption		20.1 per cent.

Both ships were put into service directly after the trials. It is now over a year since these two cargo boats have been engaged upon an exactly similar service, and it has been possible, therefore, to obtain accurate data regarding their working and comparative coal consumption.

Taking for each ship ten trips made at corresponding dates, so as to have as far as possible identical conditions of weather, loading and quality of coal used, the fuel consumption per mile worked out at 69.981 kg. (154 lbs.) for the *Garonne* without superheating; 57.228 kg. (126 lbs.) for the *Rance* with superheating.

Comparing these figures we have an economy in coal consumption in favor of the *Rance* of

$$\frac{69.981 - 57.228}{69.981} = 18.2 \text{ per cent.}$$

There was, moreover, no trouble in either engines or boilers. No leakage has occurred in the valves, which continue to bear simultaneously on both upper and lower seatings. The superheater has not required any particular attention; a constant steam temperature has been maintained, which rises only a few degrees when additional power is required of the boilers, and falls again automatically directly the engines are stopped.

In consequence of the results of the early trials of the *Rance* the Compagnie Générale Transatlantique installed Pielock superheaters and Lentz valve gears on the S. S. *Pérou*, for service to the Antilles (West Indies), and also on the intermediate cargo boat *Caroline*. The same arrangement is being adopted on the cargo boat *Honduras*, and other cases are under consideration.

These results were considered most satisfactory both by the Postal Commission and by the owners of the vessels.

A noteworthy feature is the constant temperature of the superheated steam. This temperature was taken at admission of steam to the engines by means of a Fournier recording thermometer.

It will be seen that the variations of temperature are very

small, and do not exceed 20 degrees centigrade (36 degrees Fahrenheit) from the time of starting the engines to that of running at full power, which includes also the period of cleaning the fires.

These results are very interesting as being applicable also to the use of superheated steam in large turbines on board ship. The absence of sudden changes of steam temperature in turbine engines will prevent the apparatus from being exposed to sudden expansion and contraction, the effects of which might be serious.

From the results obtained both in England and in France, superheated steam applied to reciprocating engines conduces to considerable economy without the introduction of complicated machinery necessitating additional attention. There appears to be no reason, moreover, why superheaters of a type similar to that installed on the steamship *Pérou* should not be fitted to the large turbines of ocean liners, using only a moderate amount, say, 60 degrees to 70 degrees centigrade (108 degrees to 126 degrees Fahrenheit), of superheat. From the experience gained with steam turbines on land, at least 10 to 12 per cent. fuel economy should result from this, being 1 per cent. saving for every 6 degrees centigrade (11 degrees Fahrenheit) of superheat.

Each of these vessels is to be comparable to a sister ship, working with saturated steam, in order to follow up on a larger scale the conclusive results already obtained with the *Rance* and the *Garonne*.

The steamship *Pérou* has just completed her trials. She is identical, save for the superheating and valve gear, with the steamship *Gaudeloupe*, employed on the same service, and which was completed in September, 1907.

The dimensions of these two ships are as follows:

Length	131.00 met. (429 ft. 9 in.)
Beam	15.86 met. (52 ft.)
Molded depth	10.50 met. (34 ft. 6 in.)
Load draught	6.60 met. (21 ft. 7 in.)
Gross tonnage	6,800 tons.

Each vessel is fitted with twin-screw triple-expansion three-cylinder engines of the following dimensions:

	Gaudeloupe.	*Pérou.*
Diameter of cylinders, (H.P.)	0.685 m. (27 in.)	0.685 m. (27 in.)
(M.P.)	1.092 m. (43 in.)	1.060 m. (41¾ in.)
(L.P.)	1.828 m. (72 in.)	1.828 m. (72 in.)
Stroke	1.219 m. (48 in.)	1.219 m. (48 in.)

The boiler installation of each ship comprises six cylindrical boilers having three furnaces each, and fitted with Howden's forced draft.

	Gaudeloupe.	*Pérou.*
Total grate area, sq. m	32.130 (346 sq. ft.)	32.130 (346 sq. ft.)
Boiler heating surface, sq. m.	1,255 (13,509 sq. ft.)	932.70 (10,030 sq. ft.)
Superheater heating surface, sq. m	.....	302.00 (3,260 sq. ft.)
Total heating surface, sq. m.	1,255 (13,509 sq. ft.)	1,234.70 (13,290 sq. ft.)
Working pressure	13 kg. (185 lbs. per sq. in.)	

The speed trials of both vessels, which were carried out under absolutely similar weather conditions, gave the following results:

	Gaudeloupe.	*Pérou.*
Date of trials	Sept. 9, 1907.	Feb. 6, 1908.
Steam	Saturated.	Superheated.
Boiler pressure	13 kg. (185 lbs. per sq. in.)	13 kg. (185 lbs. per sq. in.)
Temperature of steam at engines	192° C. (377.6° F.)	238° C. (460° F.)
Revolutions	88.19	88.47
I.H.P.	6,585.	6,750
Speed	16.60 knots.	16.95

(or a gain of 0.35 knot in favor of S. S. *Pérou.*)

Shipbuilders and shipowners may well turn their attention to this question, as it is likely to remove the only remaining objections to the general use of steam turbines for ocean navigation—a hope which is likely to be fufilled at no distant date.

VANADIUM IN CAST IRON.

An interesting investigation has been made by Mr. Moldenke, secretary of the American Foundry Association, regarding the possibilities of vanadium used in cast iron. Its

effects upon steel indicate its advantages when used in cast iron for railway car wheels, rolls, alkali pots, pump parts, etc.—in fact, wherever stresses are heavy and repeated, and where castings are subjected to shocks or great variations in temperature. A ferro-vanadium carrying high carbon was selected because it melted at a lower temperature and would also be cheaper for the foundryman. Varying proportions were added to the ladle of molten metal, first in lump form. As this did not give satisfaction with the small quantities of iron used at a time, the alloy was powdered before using. As vanadium, besides being a great strengthener, is also a powerful deoxidizing agent, and as the increase in strength obtained by its use might be attributed to the purification of the iron only, a further series of tests was included in which the ladle was first treated with 80 per cent. ferro-manganese in sufficient quantity to add 0.5 per cent. of manganese, and then the ferro-vanadium. In order to throw some light on the deoxidizing power of vanadium a set of tests was also made with burnt metal. All the tests indicate the value of the vanadium addition. The alloy used contained 14.67 per cent. of vanadium, 6.36 per cent. of carbon and 0.18 per cent. of silicon. While the vanadium content is comparatively low, this is a good alloy for foundry purposes, as cast iron is already high in carbon, and the silicon is too small to play an appreciable part in the results. While the attempt was made to get as nearly 0.05, 0.10 and 0.15 vanadium into the ladles of metal as possible, the analyses show that as much as two or three times this quantity remained after casting. This is due to the impossibility of accurately weighing out in the small space of time available to prevent undue cooling of the metal, when generally dealing with less metal in the ladle than had been expected or arranged for. Then, with the small quantities tried, the chances of irregular distribution were very great. A foundry with 5 or 10-ton ladles would give a better opportunity. Finally, there is the uncertainty of how much or little vanadium is oxidized. The very best results with both manganese and vanadium show

very little of the latter remaining. The results, however, are sufficient strongly to recommend the new alloy. A still better method would be to use a more powerful deoxidizer than manganese and add the vanadium on the top of it. A method which increases the breaking strength of a test bar from 2,000 pounds up to 2,500 pounds for gray iron, and 1,500 pounds up to 3,900 pounds for white iron, is sufficient to warrant further investigation on the part of every foundryman who has special problems in strength to master. It is expected to continue the investigations, making provision to keep the ladle with melted iron heated up for a fairly long period, so that better mixing of the alloy may result, and hence more accurate results can be obtained.

STEAM-ELECTRIC MARINE PROPULSION.

A paper, by Mr. William P. Durtnall, entitled, "The Generation and Electrical Transmission of Power for Marine Propulsion and Speed Regulation," was read before the Institute of Marine Engineers. Leading up through the stages of development in reciprocating engines to the adoption of steam turbines in ship propulsion, the author pointed out the few but important disadvantages involved in the direct-coupled turbine. It was necessarily designed to run at very low speed to permit the use of a propeller of high propulsive efficiency, and consequently its diameter and weight were increased to a very great extent. The blade clearance was also greater, to allow for expansion and contraction, involving higher consumption and leakage of steam. There was also the difficulty of reversing, and the increased consumption per horsepower when working below full load and speed. Tests on the *Lusitania* had shown an increase in consumption from 14.46 pounds to 26.53 pounds of steam per shaft horsepower per hour when speed was reduced from 25.4 to 15.77 knots. The ability to maneuver was of little importance with ships making long runs, but very important in short-voyage vessels and ships of war, in which

frequent starting and stopping and maneuvering in and out of harbor or in squadrons was necessary. To meet these requirements, astern turbines, of about one-third the total ahead power, and coupled to separate propeller shafts, were carried. The dead weight was therefore only partially utilized.

Full-power speed at reverse was only possible with a complete duplication of turbines, but the safety thereby secured was possibly of more importance in short-trip vessels than the increased weight involved. In order, therefore, to secure the complete success of the steam turbine for ship propulsion, means would have to be devised to allow the turbine to run at high speed and the propellers at comparatively low speed, so securing economy in both cases, and also to provide reverse motion for all shafts. After alluding to mechanical, hydraulic and compressed-air devices which had been tried and found inefficient, the method of electrical power transmission was stated to be that in which the greatest possibilities lay. This system supplied the very elements required to take advantage of high-speed turbines, saving weight and securing high economy in steam, by utilizing these to drive electric-power generators, the current so generated being used to drive low-speed motors coupled to moderate-speed large-bladed propellers of high efficiency. This method of transmission would do away with the necessity of extra turbines, shafts and propellers for reversing. The turbine generator would always run in one direction as regards speed rotation, while the motor could be reversed and efficiently run in either direction. Moreover, the direction of rotation and speed could be instantly changed to meet all conditions in practice, and this very important feature could be utilized with the utmost economy even at the lowest speeds and powers. The electric motor, as well as the turbine and generator, had also the capability of standing very severe overloads for short periods without damage. The large power conductors need only be broken under no voltage conditions, and the control could be operated from the bridge where of advantage.

The author was of opinion that for marine propulsion elec-

trical power transmission could only be successfully effected by polyphase alternating currents, with synchronous generators and squirrel-cage induction motors, on account of the low cost and weight per horsepower, absence of commutators, and high efficiency of these machines. In generators of this type the armature was a stationary closed ring with the winding embedded in slots round the inner face, inside which the field magnet revolved. The main current was generated in the stationary ring and taken off direct without any of the intermediary devices necessary when collecting current from a rotating source. The field magnet received its exciting current from a small direct-current machine mounted on the generator shaft. This current, which only amounted to about 2 per cent. of the generator output, was delivered to the revolving field through simple collecting rings. It was of importance to note the exceptionally high efficiency, light weight and low steam consumption of such combinations as converters of mechanical into electrical energy. The efficiency varied from 85 per cent. in small sets to 98 per cent. in large sizes, such as used for traction generating stations; the weight was between 35 pounds and 22 pounds per kilowatt output continuous rating, according to speed and other circumstances; while the steam consumption in 7,500-kilowatt sizes, running at 750 revolutions per minute, with 160-pound steam pressure at the stop valve, 150 degrees superheat, and exhausting into 27½-inch vacuum, was only 13.5 pounds to 14 pounds per kilowatt hour, including steam used for auxiliaries.

As regards the induction motor, its powerful starting torque, light weight, simplicity, low cost of construction, mechanical strength, durability and running characteristics, made it especially suitable for marine work. As there was no commutator and no sparking limit the output could be carried much higher per unit weight than in other machines. These motors could be built for marine work of from 1,000 to 10,000 horsepower continuous rating, weighing 35 pounds to 20 pounds per brake horsepower, with an efficiency of 93 to

97 per cent. Although polyphase motors were termed non-synchronous they always tended towards synchronism, and with squirrel-cage rotors the variation in speed from light to full load seldom exceeded 5 per cent., even in small sizes, and would probably not exceed 1 per cent. in large sizes. The main working current passed through the stationary part of the motor only, facilitating strong and reliable construction for the conductors and rendering the machine so simple as to require no skilled and very little unskilled attention with consequent low cost of maintenance.

Assuming a vessel of 4,000 horsepower fitted for electric propulsion with four propellers taking 1,000 horsepower each at 250 revolutions per minute, the motors would be polyphase induction type with squirrel-cage rotors and with stators wound for full, half and quarter speeds. The generating plant would comprise duplicate sets of turbo-alternators and exciters capable together of supplying 3,250 kilowatts, and running at 1,500 revolutions per minute, with 150 pounds steam pressure and 150 degrees Fahrenheit superheat. These alternators would be two-pole machines, and the motors would be wound for 12 poles giving a reduction of 6 to 1. For half speed a second winding would be arranged for 24 poles, giving 12 to 1 reduction with two sets of windings in parallel, and for quarter speed this winding would be arranged for 48 poles, giving 24 to 1 reduction; the synchronous speeds would thus be 250 revolutions per minute for full speed, 125 revolutions per minute for half speed, and 62½ revolutions per minute for quarter speed.

At full speed of the vessel the propellers would require 4,000 shaft horsepower, and the turbo-generators would be run in parallel delivering 3,250 kilowatts to the motors. The consumption of steam would be 13 pounds per shaft horsepower or 16 pounds per kilowatt hour, equivalent to a total consumption of 52,000 pounds of steam per hour. Compared with propeller turbines the consumption in which, under similar conditions and speed, was of the order of 22 pounds of steam per shaft horsepower or a total of 88,000 pounds per

hour for 4,000 horsepower, the saving in favor of electric transmission was no less than 41 per cent. At half speed of the vessel about 600 shaft horsepower would be required, less than half load on one turbo-generator, and this would allow the duplicate set to be shut down. Estimating the motors at 60 per cent. efficiency on half speed, the generator would deliver 740 kilowatts at its full normal speed, and take 24 pounds of steam per kilowatt hour, equivalent to 17,760 pounds per hour, as against 28,200 pounds for propeller turbines, thus saving at least 37 per cent. in steam when the vessel is moving at half speed. The weights of the electrical transmission plant in this example would be—

4,000 B.H.P. at 35 pounds per H.P. developed.....say	62½ tons.
Conductors, switch gear, &c.....	6
Two turbo-generators as above.....	70
Condenser for above..	19
Steam piping, &c.....	13
Air pump and motor.....	6
Circulating pump and motor..	7½
	184

While the weight of the direct-coupled turbines and part of the propeller shafts, tunnels, &c., including condenser, steam pipes, &c., air and circulating pumps—which would have to be arranged efficiently to deal with the 88,000 pounds of steam—would be, at least, taking the speed of the turbines at 250 revolutions per minute, 148 tons, showing that the electrical machinery would be about 25 per cent. heavier, but it is here necessary to point out that that only applies to the plant from stop-valve to propeller-shaft connections.

This comparison does not cover the saving in power and weight of boilers. On the other side of the stop valve the weight of cylindrical boilers (empty), mountings, funnel, forced-draft gear, pipe work for steam-water and exhaust, pumps, gratings, platforms, &c., is stated to be 180 pounds per indicated horsepower in reciprocating-engined ships. Taking a vessel carrying two reciprocating condensing engines developing together 4,000 brake horsepower—equal to

4,700 indicated horsepower at 85 per cent. mechanical efficiency—the weight of boiler and stokehold equipment would be 377 tons. At 16 pounds of steam per indicated horsepower, including auxiliaries and losses, the total consumption would be 75,200 pounds, or 200 pounds of steam per hour per ton of boiler-room equipment.

At this rate the boiler-room equipment on the electrically-equipped vessel, taking 52,000 pounds of steam, would be 260 tons, while for the propeller-turbined vessel the weight would be no less than 440 tons. There would thus be saving for equal propeller horsepower of 180 tons weight—that is, no less than 40 per cent. Comparing coal consumption, the author took it that, with good Welsh coal, hot feed and clean flues, it was possible to evaporate 10 pounds of water per pound of coal burnt, and at this rate a turbine-propeller vessel would burn 8,800 pounds per hour, as against 5,200 pounds by the electric-propeller vessel—a saving of 1.6 tons per hour. This represents a saving in dead weight carried of about 230 tons on a six-days' trip, with correspondingly reduced running cost.

Electrical propulsion should be of considerable interest to cargo-vessel owners, as not only could dead weight and cost be saved, but the propeller speed could be kept as low as necessary and usual in these slow boats to secure the best economy at the propeller; and it would also be possible further to reduce the propeller speed if necessary to meet practical conditions by slight increase in the weight of the motors only, the generating-plant weight remaining the same. The author pointed out that the conclusions deduced should not be considered as perfectly definite on account of the widely different conditions in practice, but the paper was chiefly put forward to lead naval architects and marine engineers to give serious consideration to the question of electric propulsion.

Reference was made to other systems proposed and a large number of lantern views and drawings exhibited. The original Hart-Durtnall continuous-current system employed series-wound generators and motors direct-coupled to six propeller

shafts, but was for small powers only. In Mr. Parsons' system the exhaust was taken from the main reciprocating engines to drive a low-pressure turbine coupled to an electric generator delivering current to a motor on the engine propeller shaft. Single-phase commutator motors had been proposed in one German and one American system; in the former case with three-phase and in the latter with single-phase generation, and in one other American proposition three-phase non-synchronous generation was included. A Swiss proposition was to use direct-coupled turbines at the vessel's full speed and a separate turbine driving an electric generator supplying current to electric motors on the main turbine shafts for reducing the vessel's speed or reversing. The reverse, however, in most of these systems was only one-third of full power. The Hart-Durtnall three-phase system, as applied to marine propulsion in conjunction with internal-combustion engines, providing a reverse and three ahead speeds with control from the bridge, was referred to, but the author pointed out that until a satisfactory internal-combustion engine of large power was produced, the steam turbine would remain the most simple, light and efficient prime mover. —"The Engineer."

SMOKELESS POWDERS.

Recent accidents have again called attention to the question of modern powders, and have once again afforded a proof that much anxiety is felt by the public in regard to these powders. The loss of the Japanese ship *Matsushima* has very generally, and without much hesitation, been attributed to a decomposition of cordite, although nothing definite is known as to the accident which led to the foundering of this ship. The explosion of the Chilian powder magazine at Batuco, in the latter end of March, was first assigned to the decomposition of French powders, which had been manufactured as far back as twenty years ago; later, however, it was discovered that the magazine contained powders of Ger-

man manufacture only; these powders were then assumed by the public to be the cause of the disaster without further inquiry.

The mere statement "decomposition," or "spontaneous combustion," of powder is both a simple and a very striking one; it cannot be said to define in any way the first cause of an explosion, but it has the curious effect of satisfying public opinion. To all thinking minds it nevertheless remains nothing but a mere statement, and it requires to be proved in each particular case. In a great number of instances, however, no attempt is made to demonstrate fully that such a statement really goes to the root of the matter.

There recently appeared in one of the best Paris daily papers an article which deals with the explosion on board the *Jena*. This article was evidently due to a technical expert, but it was written in the very discreet style which characterizes the paper and in such a way as not to contradict too openly as yet the opinion which generally obtains in regard to that explosion. It gives an explanation of the accident, and leads up to the opinion that the French "B" powder,* so generally condemned—and with many apparently good reasons, one must admit—should be exonerated in this particular case. There is, however, much that is indefinite in the article in question, because it gives only a fragmentary explanation of the cause of the disaster. It states that "* * * neither strict justice nor sound logic require us, in the case in question, to attribute the cause solely to our 'B' powders * * *." The natural conclusion that can be drawn from such a statement is that there are other explanations which better account for the occurrence, and which the French paper did not see its way clearly to publish. Such other explanations, we believe, can be gathered from the reports issued by the committees appointed by the French Parliament for investigating the causes of the disaster, which reports are criticised in very moderate terms in the article in question. It says, "* * * the publication of

* The "B" powder is the pure nitrocellulose powder used in France.

the reports issued by the committees was found to be surrounded with many difficulties. These reports put forward everything; all evidence whatsoever has been accepted without selection and without criticism, so much so that the door remains wide open to all interpretations, even to the most malignant."

We have had an opportunity of looking through the reports in question, and from these we gather the following conclusions, which, we feel confident, will not be taxed with any ill-will.

In the first place, a very striking feature of the said reports is the little weight attached to some of the evidence given. Thus, for instance, it was recorded that the aft powder magazines for the 4-inch guns, in which the explosion appears to have originated, were closed by padlocks of a most usual type, easy to open with almost any key of a size approaching the right one, or by means of a hook; the door could even be kicked open. Notwithstanding this, the reports give much significance to the statement that the magazines were locked when the explosion occurred, and proof is adduced to the effect that the keys were at the time in their right place in the commander's quarters. On the other hand, the captain of artillery, who had charge of inspecting the *Jena* after the explosion, has recorded that the most heteroclitic substances were found in the powder magazines—such as soap, paper and tobacco—adding that there might have been candles also, seeing that he found one in one of the forward magazines in which, fortunately, no explosion occurred. French naval officers state in regard to this that it is a custom on the part of the men to use the magazines as private storerooms; attempts have been made to stop the practice, but so far only with small success.

A colonel of artillery, who has under his charge the inspection of the ammunition of the fleet, stated that he saw once in the powder magazine of a ship a loaded cartridge case, fitted with a celluloid diaphragm on which a candle had been fixed—for lighting purposes, no doubt—by causing first the candle grease to run hot on the diaphragm.

The report to the Senate concludes from the above that "the fact that the magazines are used for hiding various objects illustrates a trait in naval life, but no proof can be deduced from this fact in regard to the explosion," and it sets aside, without any further investigation, the possibility of the disaster being due to an imprudent act. The report to the Chamber of Deputies, on the other hand, takes note of the possibility of imprudence, but shows a certain amount of reluctance in doing so. It states that spontaneous combustion of the "B" powder has not been proved, but lays very special stress on all points that would tend to confirm that spontaneous combustion did actually occur.

The testimony tendered by the colonel, and also by the captain of artillery above referred to, has thus been almost totally ignored, through fear, perhaps, of giving too much prominence to a deplorable practice on the part of the crews. The fact should not be lost sight of that dangerous practices are very possibly of common occurrence also in the land installations in which the powders are manufactured, for it is in human nature to forget danger when in constant contact with it.

It would appear quite clear, therefore, that the supposition of imprudent conduct on board the *Jena*, either in a black-powder ammunition hold or in one containing "B" powder, should not so summarily be dismissed. The possible consequences of an imprudence should, on the contrary, have been weighed in the light of the actual occurrence. By supposing an inconsiderate act to have been committed in a smokeless-powder hold, the result, as regards consequences, would be the same as that due to spontaneous ignition, for it is quite clear that, whether spontaneous or not, the inflammation of powders can only propagate in one and the same direction after a first ignition, when all charges are stored in the same way. The French journal sums up the occurrence very clearly; it states: "The characteristic feature of all the accidents, without exception, which have happened on board ship up to the disaster on the *Jena*, by reason of the inflam-

mation of "B" powder, is the slow and gradual development of combustion. There have always elapsed several minutes between the first appearance of flames and the arrival of assistance, and in no case has the first manifestation of the accident had any directly serious results. Men caught in the midst of the deflagration have been able to escape easily without getting seriously hurt. The experience was similar in the course of the test carried out by the Gavre Committee, at Lorient. This test was made with a view to ascertain what would happen in a powder magazine of the type of those of the *Jena*, in which a charge was ignited in the midst of others. After the first ignition, followed by the projection of alternately increasing and decreasing flames and jets of gases, five minutes elapsed before the first explosion took place."

This characteristic feature does not appear to have received sufficient attention on the part of the examiners; it is not given prominence in their reports, though it is confirmed by many witnesses. When a similar accident occurred on board the *Forbin*, the officers hesitated for some time with regard to the place whence the first flames issued, and had, besides, to improvise a fire service, a new crew having only just come on board; notwithstanding the consequent delay, the flames were extinguished before any explosion occurred. In the case of the *Duperré*, again, the first signs of inflammation were noticed at 11·45 in a port magazine for the 13.3-inch guns; no shattering effects occurred, although the fire was not under control until 12·15, and all danger only disappeared as late as 12·35. The five minutes which elapsed in the case of the Gavre experiment above alluded to may evidently be taken as a minimum, seeing that artificial ignition was carried out at the base of a cartridge, the black priming powder being lighted first. There is no evidence to show that in an instance of accidental ignition, or in a case of decomposition, ignition always first takes place at that same part of a cartridge; the experiment, therefore, introduced again another assumption, and one corresponding to a most dangerous eventuality, but the

experiment afforded no proof whatever that the occurrence could take place spontaneously; combustion also developed slowly and progressively in this experiment.

The events on board the *Jena* were in no way progressive, as in the cases we refer to; they commenced by a shattering explosion, which killed a number of men on the spot. The whole of the evidence is practically unanimous on this point. Only one witness gave it to be understood that flames were seen for two or three minutes before the first explosion, and the greatest importance has been given to this single evidence, particularly in the report to the Senate. But, on the other hand, it should be remembered that sailors were near and above the holds at the time; the hoist hatchways from the magazines opened directly on the deck on which members of the crew were dismounting a 4-inch gun. It is therefore absolutely incredible, under such conditions, that flames could have been emitted during two or three minutes without an alarm having been given. The officers killed in their cabins were not simply suffocated; they were crushed to death by the explosion. All tends, therefore, to prove that a shattering explosion occurred without any warning whatever, and this is confirmed by Admiral Manceron, a survivor of the disaster.

It is difficult, therefore, to entertain the supposition that powder "B" became ignited, either spontaneously or by reason of an imprudence, for this supposition does not fit in with the facts ascertained in the case of the *Jena*, or with those in the case of other ships. In order to maintain this supposition it would be necessary to surmise still further—namely, that the "B" powder behaved in the case of the *Jena* in a new and unknown manner.

If, on the other hand, the possibility of an imprudence be considered, which led to the ignition of black powder in one of the magazines containing this latter powder, the accident becomes easily explicable without it being necessary to entertain any consecutive suppositions. Black powder always explodes with shattering effect, and in the case of the *Jena* this

would thoroughly explain why so many men were killed almost instantaneously. Ignition was propagated to the neighboring smokeless-powder magazines, then to the other black-powder magazine, this explaining how a witness, after having escaped the first explosion and placed himself out of further danger, could see flames issuing from the ship during two or three minutes.

As a conclusion, we think that the hypothesis of an imprudence committed in one of the black-powder magazines of the *Jena* is that which the best withstands impartial consideration, and we add that it might very well be that the French have depreciated more than they really had cause to do their smokeless powder. Following the disaster in the French ship, which we all deplore, our own Navy gave special attention to the matter of our cordites and their conservation; this is certainly a most excellent measure.—"Engineering."

FRENCH WARSHIP CONSTRUCTION.

It is no exaggeration to say that among the questions connected with naval architecture none has so great an importance as that which relates to the time occupied by the construction of ships. This statement forms the opening of a paper read by Mr. C. Ferrand, Ingénieur en Chef de la Marine, at the meeting of the Association Technique Maritime, held in Paris, on May 6 last, in which he compares the activity displayed in British and German naval yards with the dilatoriness that generally obtains in France. In pointing out the fact that France takes five years to build the ships which Germany can build in three years and England in two, Mr. Ferrand adds that the five years he mentions apply to the French Government yards; the private yards, when they are not hampered by Government red tapeism, can build quite as rapidly as the German yards.

In France the time reckoned for the construction of a ship counts from the date upon which the order for laying it down

is issued by the Admiralty to that of the entrance of the ship into actual service; in England and Germany the time counts from the actual laying down to the completion—that is to say, to the commencement of the trials. Now in France there is a long interval between the order and the actual laying of the keel, and an approximately equal interval is taken up by the trials. The author illustrated this by the following figures:

	Date of order.	Date of laying keel.	Interval.
République	June 28, 1901	Dec. 2, 1901	5 months
Démocratie	April 5, 1902	March 1, 1903	13 months
J. Michelet	April 5, 1902	June 1, 1904	26 months
Waldeck-Rousseau	July 31, 1905	June 16, 1906	11 months
Danton	May 8, 1906	April, 1908	23 months

	Commencement of preliminary trials.	Entrance into effective service.	Interval.
Charlemagne	April 2, 1898	Sept. 6, 1899	17 months
Suffren	Aug. 30, 1902	Feb. 5. 1904	18 months
Gambetta	Feb. 22, 1904	July 21, 1905	17 months
République	July 10, 1906	Jan. 1, 1907	5 months

The above indications would show, according to Mr. Ferrand, that when it is stated that France takes five years to build a ship, while England built the *Dreadnought* in one year only, the comparison is not a fair one, seeing that the *Dreadnought* was commenced in January, 1906, and completed in January, 1907; but the order for the ship was given before midsummer, 1905, and her trials took up more than half of 1907. Hence, by counting the time as it is counted in France, over twenty-four months, and not one year only, were occupied by the *Dreadnought*—a result which is sufficiently remarkable in itself and does not require any exaggeration. Hence, also, the comparison, five years taken by France, three years by Germany and two by England for building a battleship, given by the author of the paper.

	1906.	1907.	1908.	1909.	1910.	1911.
	£	£	£	£	£	£
Danton (building in Government yard) ...	66,612	209,602	356,864	526,059	539,400	234,400
Voltaire (building in private yard)	12,253	183,303	271,520	480,790	624,320	499,640

The financial question is at the root of the whole matter in regard to delay. An example of the distribution of expenditure, according to the budget of the present year, is given in the paper, in the table above.

If the French navy succeeded in building in three years the ships the order for which was given out on May 8, 1906, these ships would have to be paid on the budgets for 1906, 1907, 1908 and 1909; by dividing the cost over the four years instead of over the six the budget for the Navy would have to be increased by £960,000 in 1906, by £2,000,000 in 1907 and by £1,800,000 in 1908. This cannot be done, and the navy offices have to wait until 1909 and 1910 before there will be allotted to shipbuilding yearly payments of such amounts as will allow of really active work being carried out. Seeing that the regulations in regard to accounts require that the work effected in any one year must be paid out of the funds provided for that same year, it is necessary to proceed rapidly with the ships ordered in 1906 only during the years 1909 and 1910, at a time when they ought to be placed in active service. The situation is such that if the yards built quicker than is prescribed, effective measures would have to be taken to delay them. The Central Naval Offices in Paris, through lack of available funds, have been obliged on various occasions to restrict the activity of the yards; this, adds the author, is ancient history, but the fact, nevertheless, remains that the yards are compelled to proceed slowly, and this has favored, as it were, the numerous existing causes which make for delay.

As a remedy, the author would propose to include the construction of new ships in a special extraordinary budget, as is done in Germany, and when the construction of a new unit had been voted by Parliament, the necessary subsidies for completing it would run no risk of being declined by Parliament; the necessary funds would be considered as placed at the disposal of the Navy, which would then be able to utilize them to the best possible advantage. With the present practice, however, the Navy Department draws out programs in

which are provided amounts for forthcoming years much in excess of those for the year under immediate consideration; when the following years are reached, no further consideration is given to the provisions formerly drawn up. The aim of the Minister of Finance is that expenditure be maintained low; the difficulty is met by carrying the provisions forward by one year or two, and the result of this mode of operation is to delay more and more the fulfilment of a program.

In dealing with the manner in which the building of a ship should proceed, the author states the very evident fact that the drawings should be got out and the contracts for the material and machinery passed in a definite order corresponding to the rational progression of operations at the naval yard. Thus, if it takes twenty months to build the turrets, twenty-four months to build the engines, and twelve months to build the hull, it is clear that the engines should be put in hand first, then the turrets, and lastly the hull. The French navy does not proceed on these rational lines; it gives out the order for the hull first, then a long discussion is started on the type of turrets to be used, and these are ordered at a time when they ought to be completed. The same takes place in regard to to the boilers; a selection among the various types of boilers raises such an amount of discussion that the Secretary of State for the Navy postpones a decision and an order up to the last moment, hoping, probably, to bequeath to his successor in office the difficulties and the responsibility of a selection. The following dates speak for themselves:

	Democratie.	*E. Quinet.*
Date of order for the ship	April 5, 1902	Aug. 27, 1904
Date of laying the keel	March 1, 1903	Nov. 6, 1905
Date of contract for 12-inch turrets	Aug. 4, 1903	.....
Date of contract for 7.66-inch turrets	July 13, 1904	July 16, 1907
Date of contract for boilers	June 6, 1904	Aug. 3, 1906

In this connection it should be borne in mind that the naval yards require to keep their *personnel* regularly employed, with the result that it becomes necessary at a given time to put in hand a fresh unit to occupy the labor which gradually

becomes available with the completion of a former unit. In order to meet this the Secretary of State would have at the right moment to take steps to obtain from Parliament the authorization for laying down a fresh unit, for which the necessary credits should be made available and the necessary plans completed. But the situation at the naval yards is never so met. The Navy, it is true, endeavors to approach Parliament in good time, but it is in Parliament that difficulties occur; thus the program for 1906 was voted at the commencement of the year, but was placed on the carpet again by a series of questions at the period when the orders were about to be given out, and this led the Secretary of State to obtain a fresh vote from Parliament before he finally signed the contracts.

The original outline drawings are discussed and altered under the varying influences of Parliament and the Navy, and up to the last moment schemes are dealt with, and not final data. Then arrives a day on which the Secretary for State gives out the order for laying down the unit; but this is done with the sole object of closing discussion and making it possible to start on the detail études and drawings. It is therefore a fallacy to date the actual commencement of a ship from that day. The order for the ship being given, the Admiralty immediately proceeds with the business reckoned as the most urgent, this being generally to distribute work to the naval yards, the workmen in which, through lack of orders, are on the point of standing with folded arms; hence the hull is put in hand much before the engines, boilers, turrets, &c. The work on the hull forms a most unsatisfactory operation, nothing being yet known as to the dimensions and the position of any of the essential parts of the ship. It is carried out without order or method, simply to occupy the men; and large portions of the ship are built which, it is well known, will have to be taken to pieces later on. Owing to alterations in the plans, several of the ships are designed twice over in course of construction, and there is not the least doubt, says the author, that should the French Admiralty, using the same

methods, entrust one of the largest British shipbuilding yards with the mere construction of a battleship of, say, the *Danton* class, its delivery would not take place sooner than if built in a French yard.

He explains at length the causes which make for delay; these can be summarized in a lack of proper understanding between the designing and managing departments and those in charge of execution. Red-tapeism, the interchanging of papers and visas for every trifling detail, also play a large part in causing delay. Thus, when a naval yard or a private shipbuilding firm raises a point of detail in the construction of a ship, the reply from the Ministry takes three months in the ordinary course; should discussions, however, be raised, the loss of time is increased to six months, and occasionally to one year. The differences of opinion entertained by the Ministers who succeed each other at the head of the Navy have also been the cause of serious delay in the increase in number of French fighting units. It occurs also that orders for fresh units are given out at a time when the men are fully occupied with work on hand. This method of proceeding, says Mr. Ferrand, is preferable to that referred to above, but it has the disadvantage of increasing the nominal time taken for construction.

Interesting dates are given to illustrate the delay experienced in the case of various ships of recent construction, and the very surprising statement is made that when a Government yard has received from the Ministry the order for laying down a unit, the yard has to ask the Ministry to sanction the purchase of the required plates for that same unit.

The author also reviewed the situation as regards private companies. Whereas it takes a fortnight in England to obtain plates and sections, it takes in France three or four months. The French works require six to eight months to supply steel castings of simple shape; a sternpost cannot be delivered under a year. Such unfavorable conditions are partly explained by the hesitancy and absence of method which rule in the Ministry of Marine. If the latter had defi-

nite programs, and worked up to them, the industrial concerns of the country would be less timorous; they would, if need be, increase their means of production, with a view to manufacture in larger quantities and more rapidly.

The inspection of the material in course of manufacture is another cause for delay. It is essential, of course, that material of first quality only should be used in ship construction, but now that the manufacture of steel has reached such a high degree of perfection, and that the heroic ages are past, is it necessary, asks the author of the paper, that active means should be taken to prevent the putting in place of a plate or a rivet when the manufacture of the metal used for these has not been witnessed by the naval inspector? Time is also lost in armor-plate tests. When a plate for testing is selected from a lot, a suitable backing, similar to the ship's side, has to be built up, and several months are spent in correspondence, work and tests for each separate plate. The Government engineers, moreover, do not concern themselves sufficiently with setting out simple shapes, easy to manufacture, and very often the armor plates, in particular, are designed in total unmindfulness of the difficulty of execution.

The trials of ships also take up too much time; the trials of all the numerous auxiliary engines are often very laborious. For the latter class of engines, competition has been carried to excess, with a view, perhaps, to save £4,000 first cost on a ship of £2,000,000; this saving is more than compensated for by more lengthy trials and difficulties in actual service. The time taken by the trials is further increased by the practice, now very general, of ordering from different builders the boilers, the engines and other main apparatus, all of which require independent tests.

Some of the remedies advocated are stated in the foregoing. The paper concludes by detailing them; they are, briefly, decentralization in some departments, centralization in others, giving the right persons the required authority, and less red tape. No far-reaching reforms are necessary, says the author, but only measures of detail, such as a minister can obtain by a decree.

Considering the decrease in the number of men employed in the naval yards, the reduction of the working hours to eight and the state of mind of the men, the author very greatly questions whether the 18,000-ton battleships now in progress will be completed in the four years originally stipulated. —" Engineering."

NEW BRAZILIAN BATTLESHIPS AND THEIR ARMOR.

The admirable reliability of modern armor, as manufactured on the Krupp cemented process, has had no more convincing demonstration than that published on page 825. The plate, after an attack equal to 2,900 foot-tons per ton of plate, showed at the four points of impact a penetration of the surface for a depth of only about 2½ inches from the face. This plate is part of the armor manufactured at the Openshaw Works, at Manchester, of Sir W. G. Armstrong, Whitworth & Co., Limited, for the Brazilian battleship *Minas Geraes* and others now in course of construction at the Elswick Works of the firm and at Messrs. Vickers Sons & Maxim, Limited, Barrow-in-Furness, and the result achieved in resistance is consonant with the general excellence in all qualities of design anticipated in these ships.

Much has been said regarding these battleships, and before dealing with the details of the armor-plate trials some particulars of the vessels may be given. Three such vessels have been ordered—two from the Elswick Works and one from the Naval Construction Works of Messrs. Vickers Sons & Maxim, Limited, where also the machinery of all three will be constructed. The vessels belong to what is known as the "*Dreadnought*" era, but mark a decided advance in respect of gun-power of the vessels hitherto ordered, as, instead of having ten 12-inch guns, these new ships mount twelve such weapons, and all are of 50 calibers in length. Moreover, while the *Dreadnought* can bring only eight guns to bear on either broadside, the Brazilian ships can train all of the twelve guns on either broadside. This great advantage has been

achieved by a difference in the disposition of the guns. In the waist of the ship are two twin-gun barbettes, one on the port and the other on the starboard side, but arranged *en echelon*, so that the guns can be used on either broadside. Forward and aft there are the usual center-line barbettes, and behind each of these centrally-situated barbettes there is a second, in which the guns are arranged at a higher level, and sufficiently to the rear to enable them to fire over the weapons in the barbette in front without any probability of interference from flash. Thus it is possible to fire eight guns ahead, eight guns astern and twelve on either broadside; and as the guns are of exceptional power separately, the result is to make these Brazilian ships of immense offensive power. The vessels are to displace 19,250 tons. Reciprocating engines are being adopted for at least the first two of the ships, and, as regards the third, it has not yet been decided as to whether turbines will be preferred. The speed is to be 21 knots.

The armor protection of the ships is admirably arranged, and the side plating will extend from bow to stern. The greater part will be armor weighing 360 pounds per square foot, and for the *Minas Geraes* this armor has been constructed at the Openshaw Works. It was prescribed in the contract that a plate should be selected by Admiral Duarte Huet de Bacellar, President of the Commission of the Brazilian navy, and a recognized authority on ballistics, who has been closely associated with the construction of the new vessels. The plate thus selected was mounted in a cell at the company's proving ground at Ridsdale. The backing consisted of 2-foot oak balks, laid in two layers transversely to each other, with a skin plating at the rear, so as to represent as far as possible the actual condition on the ship. The plate, with its backing, was secured by ten bolts, while to the rear there were heavy stop plates, as is usual. The trials took place on July 21.

It was specified that the attack should be by a 9.2-inch 40-caliber gun, using 380-pound projectiles made by Messrs.

Thomas Firth & Sons, Limited, of Sheffield, according to the British standard service pattern, and that the charge should be such as to give a striking velocity of not less than 1,850 feet per second, and as close an approximation to 1,900 feet as is possible, the range being 277 feet.

The results are set out in the annexed table, recording the ballistics for each round.

RESULTS OF TESTS OF 380-POUND K. C. ARMOR PLATE FOR THE BRAZILIAN BATTLESHIP *MINAS GERAES*.

No. of round.	Nature of projectile and weight.		Charge.		Striking velocity.	Energy.	Remarks.
	Armor-piercing shot.		Cordite.				
	lbs.	*oz.*	*lbs.*	*oz.*	*foot-secs.*	*foot-tons.*	
1 *a*	381	8	55	4	1,895	9,497	No cracks developed.
2 *b*	380	12	53	12	1,877	9,288	No cracks developed.
3 *c*	381	12	53	12	1,872	9,273	No cracks developed.
4 *d*	380	8	58	12	1,977	10,312	Fired to measure further resistance of plate; no cracking resulted.

a Shot broke up; caused very slight penetration; point remained embedded until third round.
b Similar result to round 1.
c Similar result to round 1. Point of round 2 shaken out.
d Similar to round 3, with slightly increased penetration.

The plate was accepted as result of first three rounds.

The first round resulted in the development of a striking energy of 9,497 foot-tons, and the projectile, which weighed 381 pounds 8 ounces, was broken up, the point alone remaining embedded to a very slight extent; but when the third round was fired this point fell out, and it was subsequently recovered, very much abraded. The second round was, so far as its results are concerned, an exact repetition of the first, the penetration of the point being again about 2 inches. The third round, which struck the lower central part of the plate, had the effect of releasing the points of the first and second shots. The plate was then officially accepted as complying with all the conditions of the contract. The splinters from around the points of impact were very thin and had a razor-like hardness. In no case was the penetration more than 2½

inches, and there was no evidence of cracking. The back of the plate was, on later examination, found to show no breakage of surface.

The condition of the plate suggested the idea of firing an extra round with a larger charge, increasing the velocity by nearly 100 feet per second to 1,977 feet, and the energy developed to 10,312 foot-tons. The result proved very satisfactory. There was only slightly increased penetration, and, notwithstanding the close proximity of the point of impact to that of previous rounds, there was little splintering and no cracking. It was obvious that the plate could have withstood an attack with a striking velocity of fully 2,000 feet per second in the case of all three specified rounds. The heads of the projectiles Nos. 3 and 4 remained embedded in the plate, while the small pits left by the points which were shaken out from Nos. 1 and 2 shots show the limited extent of penetration.

The results, as we have said, are very interesting and indicative of the great progress made in the efficiency of armor. In this case it will be noted that the bore of the gun slightly exceeded the thickness of the plate, so that the test was rather more severe than usual. We are the more pleased at being able to give these results in view of the modern tendency towards secrecy in such matters.—"Engineering."

TORPEDO AND ANTI-TORPEDO ARMAMENT.

In a reply to a parliamentary question as to whether it was proposed to replace the 12-pounder guns in H. M. S. *Dreadnought* by weapons of larger caliber, the Secretary of the Admiralty states that the 12-pounder is "not considered inadequate" as a protection against torpedo attack. We may smile at so ingenious a reply, which in a given number of words gives the minimum of information and at the same time evades the question; but while we smile we may at the same time ask what is adequate protection against torpedo attack? We shall endeavor shortly to set out the main con-

siderations on which the answer depends. There has been for some time past a large proportion of naval opinion in favor of increased armament in destroyers, and in the later vessels of the *Tribal* class and in the *Swift* the 4-inch gun takes the place of the 12-pounder, which had hitherto been the largest size fitted. No doubt the difficulty of making the change earlier has been due to there being no modern Q.F. between the 12-pounder and the 4.7, and also to some extent from the consideration of stability involved by mounting heavier guns in the position which they occupy high above the water line in vessels of small beam.

It has been assumed by some that the change having been made in destroyers, it should apply equally to the anti-torpedo armament of the capital ship, and this, no doubt, is the assumption which prompted the question to which we referred at the beginning. Such an assumption, however, shows only a partial appreciation of the problem. The function of the destroyer is to catch and destroy the enemy's torpedo boats, and to make sure of its object it must have both speed and a suitable armament. For the latter, we may say that what is required is the equivalent of the man-stopping bullet. The torpedo boat which may conceivably be approaching within torpedo range of the battle fleet must be stopped and rendered useless at the earliest possible moment, consequently the harder and the fewer the hits which the destroyer requires to give to ensure this the better. On account of the greatly superior speed of the destroyer over the torpedo boat it is practically certain that such actions will be fought at close range, and a high muzzle velocity is not, therefore, essential in the gun carried by the destroyer, while the weight of the shell is of the utmost importance. Thus a short 4-inch gun with a comparatively low muzzle velocity, throwing a 25-pound shell, is very greatly to be preferred for a destroyer to the long 12-pounder.

But while it is the function of the destroyer to get to close quarters with the enemy's torpedo boat, it is the duty of the battleship to keep the torpedo boat, if possible, out of torpedo

range, and the anti-torpedo gun must therefore have a long range, that is, high muzzle velocity. Once the torpedo boat is sighted it becomes a question of time how soon it can be disabled, and as the range will generally be fairly long, it is obvious that the greater the number of guns which can be trained on the attacking boat, and the greater the rapidity of fire, the greater will be the chance of repelling the attack. Of course, the larger the gun the better for the purpose, provided that the essentials of number which can be trained on any one bearing, of muzzle velocity, and of rapidity of fire, are maintained. The 4-inch gun now being fitted in destroyers, which is comparatively short, does not comply with the range requirements, and it is obvious that if the muzzle velocity, which depends mainly upon the number of calibers in the length, is to be maintained, much greater space will be required for a 4-inch than for a 12-pounder. The main armament of a battleship is the first consideration, and the anti-torpedo armament must take up what space can be allotted for it after the main armament has been provided for, and it must also take such positions as will not interfere with the working at all angles of training of the heavy guns.

The *Dreadnought* mounts twenty-seven 12-pounders, and probably not more than ten of these could be trained on any one bearing, and if we consider the possibility of a single ship being attacked by several boats simultaneously, it will be realized that this number may not be more than adequate to prevent her being torpedoed. It is also possible that an equal number of suitable 4-inch could not be fitted without interfering with the main armament, either in the present *Dreadnought*, or by a rearrangement of the guns in a new design. In this connection it is interesting to note that the *Invincibles*, which are considerably longer than the *Dreadnought* and carry a smaller main armament, and which therefore have presumably considerably more space available for their anti-torpedo armament, are to carry sixteen 4-inch guns of a new pattern. Had it been possible efficiently to mount a greater number, no doubt the comparatively small addition to the

displacement would have been accepted. The advantages and disadvantages of a certain number of 12-pounders, or so many less 4-inch, have, no doubt, been fully weighed, but their lordships evidently intend to keep their opinion to themselves at present, and hence the delightfully turned phrase that the 12-pounder "is not considered inadequate."—"The Engineer."

SHIPS.

ARGENTINE.

The Argentine Government are making inquiries for new destroyers. Four ocean-going vessels of 650 tons and 27 knots, and eight of 450 tons and 30 knots, are projected. Inquiries have also been made for battleships of exceptional size.

Armored Gunboat Parana.—Recently Sir W. G. Armstrong, Whitworth & Co., Limited, launched from their Elswick shipyard the armored gunboat *Parana*, which is being built for the Argentine Government. In consequence of the shipwrights being on strike, the launching ways had been laid by apprentices under the direction of the foreman shipwright, and it is satisfactory to note that the vessel entered the water without any hitch whatever. Captain de Montes had charge of the construction of the ship.

From an interesting speech delivered by Sir Andrew Noble shortly after the launching ceremony, it appears that the *Parana* has a length over all of 250 feet; length between perpendiculars, 240 feet; breadth, molded, 22 feet 3 inches; depth, molded, 14 feet; mean draught, 7 feet 6 inches, and displacement about 1,000 tons. Her armament will include two 6-inch howitzers, six 3-inch, 50-caliber, Q. F. guns, four 75-millimeter, 12-caliber land guns, and eight rifle-caliber machine guns. The *Parana* and her sister ship, the *Rosario*, besides having powerful armaments, are well protected for vessels of their class, and, in addition to other protection, they will have an armor belt extending over the machinery spaces and magazines. It is believed that combining, as they do, such good offensive and defensive qualities, they will be the most powerful vessels of their kind in existence. The *Parana* is the fourth vessel the Elswick firm has constructed for the

Argentine navy, the first being the *25 de Mayo*, a cruiser of over 3,000 tons, the second and third being the *Nueve de Julio*, of 3,500 tons, and the cruiser *Buenos Ayres*, of 4,800 tons. In addition to these vessels, the allied firm of Ansaldo, Armstrong & Co. have during recent years constructed the armored cruisers *Garibaldi* and *Pueyrredon*, each having a displacement of about 6,750 tons. The armament for all these vessels was supplied from Elswick, and in addition, the firm has supplied the armament for other vessels of the Argentine navy, including the first-class cruisers *General Belgrano* and *General San Martin*. The *Parana* has been specially designed for river service. Her draught is small, and she will have good maneuvering power. Special arrangements have been made to guard against the extreme heat to which the gunboats will sometimes be exposed on the great rivers of the Argentine, and ample provision has been made for keeping the ship cool throughout, and especially with a view to maintaining a comfortable temperature in the magazines and machinery spaces.

AUSTRIA.

The Austrian Government is building a special type of scout of 3,500 tons and 26½ knots designed speed. The length is to be 425 feet and the beam 42 feet. The Stabilimento Technico, of Trieste, is responsible for hull and machinery, of which the latter includes turbines and sixteen Yarrow boilers.

AUSTRO-HUNGARY.

Two Shallow-draught Gunboats have been lately constructed to the order of the Austro-Hungarian Government by Messrs. Yarrow & Co., Limited, at their Poplar Works. They are intended for special river service on the Danube, and are being forwarded overland from Hamburg to Buda-Pesth. The dimensions are 60 feet in length by 9 feet beam. The hulls are of steel, galvanized, and all the vulnerable parts are protected by chrome-steel protection plating, proof

against the Lee-Metford rifle point blank at short range. The protection plating covers the entire machinery space, the revolving gun tower, the conning tower forward and the petrol tank at the stern. The petrol tank apparently forms part of the hull, but in reality it is quite a separate structure from the hull proper; so that in the event of this tank being damaged by shot or otherwise it is impossible for any of the petrol to find its way into the vessel.

The propelling machinery consists of five 70-brake-horse-power Yarrow-Napier four-cylinder petrol motors, arranged to drive three shafts. The two wing shafts are each driven by two sets of engines placed in tandem, having four cylinders to each set, *i. e.*, eight cylinders for each wing shaft, while the central shaft is driven by a single set of four cylinders. The dimensions of the cylinders are 6½ inches in diameter, with a 6-inch stroke. The reversing gear is in connection with the central single engine only. A special arrangement for inducing the circulation of air through the engine room is provided by means of a centrifugal fan, so as to avoid the possibility of any accumulation of explosive gases. The forward compartment is fitted up for the sleeping accommodation of two men, and the after compartment for that of four men, lockers, &c., being also provided. There is a mast and crow's nest fitted amidships, so as to secure a good range of vision over the high banks of the river, and a revolving gun tower will be placed aft, on which will be mounted a small gun. This gun tower is to be fitted in Austria by the authorities there.

Special arrangements have been made to prevent the vibration common to some of these fast lightly-constructed vessels, and in this respect the trial was considered by the Austrian authorities as a great success, as both vessels were found to be extraordinarily steady. Steering stations are provided in duplicate, one forward in the conning tower and one amidships.

The official trials of these two little vessels took place on the River Thames, over the Admiralty measured mile, on

June 22 and 23, the Austro-Hungarian Government being represented by Captain N. Von Wawel-Louis, Naval Attaché, Herr H. Wagner, Chief Constructor, Herr J. Seifridsberger, Chief Engineer, and Lieutenant Hoppe. The trials consisted of a continuous run of one hour, carrying a load of 3 tons, and the mean speed of the two vessels was found to be 22¼ knots.

A subsequent trial took place, of one hour's duration, to test the radius of action, and it was found that at 11 knots the radius of action was at least three times as much as it would have been had the vessel been propelled by steam.

A special condition in connection with these vessels was the limit of draught, which, to the bottom of the screw, was 2 feet 8 inches.

These two gunboats are very similar to the one purchased two years ago by the British Admiralty, now called *Mercury II*, which attained, when light, a speed of between 25 and 26 knots. This little vessel has been in commission for the last two years, most successfully.—"Engineering."

BRAZIL.

Minas Geraes.—This very important vessel has been launched from the Elswick Shipyard of Sir W. G. Armstrong, Whitworth & Co., Limited, Newcastle-upon-Tyne. She is the first of three battleships recently ordered from the firm for the Brazilian navy and is by far the most important vessel that has ever flown the Brazilian flag. She has a greater displacement and a larger armament than any battleship hitherto launched in this country for our own or any other Navy.

For some years it has been the intention of the Brazilian Government to completely reorganize their Navy, which, as regards many of their ships, had become somewhat obsolete. The original scheme was to build three battleships of moderate dimensions, together with some armored cruisers, fast scouts and torpedo-boat destroyers, and Messrs. Armstrong were entrusted in a great measure with drawing up proposals for the whole scheme of reorganization. It was largely, no

doubt, owing to the experience obtained from the Russo-Japanese War, followed by the advent of our own *Dreadnought*, that it was finally decided by the Brazilian authorities, in consultation with the firm, to adopt a type of vessel combining a most powerful armament together with good armor protection and a high speed, all of them qualities which necessarily involve a vessel of very large displacement in order to embody such a combination. The *Minas Geraes* was, therefore, designed by Mr. J. R. Perrett, the head of the shipbuilding department of Elswick, to meet the Brazilian demands, which, in addition to the main features, involve many special requirements to suit the Brazilian service, the vessels of that Navy being throughout the greater part of the year in a tropical climate.

In view of the decision arrived at by the Brazilian Government to place the order for the construction of these three larger battleships, it was decided, for the present, not to build the armored cruisers proposed in the original scheme. There are, however, under construction at the Elswick Shipyard of the Armstrong Company two very fast scouts, of 3,000 tons displacement each.

The main features of the *Minas Geraes* are as follows, viz: Length 500 feet, breadth 83 feet, with a displacement approaching 19,500 tons on the normal draught of 25 feet. The guaranteed speed of the vessel is 21 knots, and the bunker capacity is 2,000 tons, which is sufficient to give a very large radius of action at a moderate speed. Oil fuel is arranged for in addition to coal.

The machinery and boilers of the *Minas Geraes*, and also of her sister ship building at Barrow-in-Furness, are being constructed by Messrs. Vickers Sons & Maxim, Limited. The engines are of the ordinary reciprocating type, this arrangement being considered most suitable for the Brazilian service.

The armor, which is being manufactured at Messrs. Armstrong, Whitworth & Co.'s works at Openshaw, Manchester, is of the Krupp cemented quality, with a main water-line

belt 9 inches in thickness, slightly tapered fore and aft. This belt is carried to the height of the upper deck over the citadel, thus affording complete protection to all the barbettes, machinery, boilers, magazines, etc.; and over the citadel the upper deck itself is thickened to form a protective deck, in addition to the protective deck proper, which runs all fore and aft in the usual manner.

The armament consists of twelve 12-inch guns twin mounted in six barbettes. Four of these barbettes are carried on the upper-deck level, one forward and one aft on the center line of the ship, and one amidships on each broadside. In addition, two barbettes are carried at a higher level on the center line, firing over the other two center-line barbettes, special arrangements being adopted to protect the latter when the upper guns are firing over them. This arrangement admits of eight 12-inch guns being fired ahead or astern, or ten 12-inch guns on either broadside at one time. The secondary armament consists of twenty-two 4.7-inch guns, fourteen of which are carried on the main deck behind the protection of the citadel armor, the other eight being carried in well-protected positions at a higher level. There are, in addition, some 3-pounders and smaller guns.

It is hardly necessary to add that the vessel will have a very complete outfit and equipment of every kind, similar to that of first-class battleships in our own and other navies; very special arrangements have been made for ventilating the ship throughout, and more particularly in the magazines and machinery spaces, for which special cooling apparatus has been provided.

This and the other vessels now building in Great Britian for the Brazilian navy, including the torpedo-boat destroyers which are in course of construction at the works of Messrs. Yarrow & Co., Glasgow, are all being constructed under the supervision of a Commission sent over by the Brazilian Government, the present very able Chief of this Commission being His Excellency Admiral Duarte Huet de Bacellar, who has the assistance of numerous Brazilian officers, who were present

at the launching. The vessel's launching weight was over 9,000 tons.

For those who are specially interested in the *Minas Geraes*, we may mention that a very fine model of her is to be seen in the exhibits of Messrs. Armstrong, Whitworth & Co., Limited, at present on view at the Franco-British Exhibition.—"Engineering."

BULGARIA.

Bulgarian Torpedo Boats.—Bulgaria owned, up to quite recently, one single warship, the *Nadiejda*, a small 715-ton cruiser, built in 1898 by the Chantiers et Ateliers de la Gironde, Bordeaux; the *Nadiejda*, besides undertaking cruising duties, serves also as a training ship. Her principal dimensions are as follow: Length, 70 m. (228 feet 8 inches); Breadth, 8.16 m. (26 feet 9 inches); Draught, 3.20 m. (10 feet 6 inches.) She is propelled by two 2,600-horsepower engines, supplied with steam by Lagrafel et d'Allest boilers; her speed is 18.5 knots. Her armament consists of two 100-millimeter (3.93-inch), two 65-millimeter (2.56-inch), and two 47-millimeter (1.85-inch), quick-firing guns, and two torpedo-launching tubes. Her officers are of French nationality, and, under their command, she passed through the Dardanelles as a steam yacht for the reigning Prince, in which capacity she is also frequently used.

In 1903 the Bulgarian Government decided to reorganize their military forces, and included in their program a scheme for defending their coasts on the Black Sea. This scheme provides for the construction of a flotilla of torpedo boats of 90 to 100 tons, running at 26 knots speed and having a large radius of action.

The situation of Bulgaria, from both a political and an economical standpoint, rendered the carrying out of the scheme somewhat difficult, for, owing to the Treaty of Berlin, the craft could not be delivered under steam through the Dardanelles, and the industrial resources of the country did not allow of the boats being built completely on the spot.

Owing, further, to the ice floes, which frequently close the port of Varna during part of the winter, provision had to be made to haul up the craft from the river when cold weather set in.

The Bulgarian Government invited tenders for the work and decided, in 1904 and 1906, to order two sets of three torpedo boats each from Messrs. Schneider & Co., whose designs best met the desiderata put forward. The program included the construction of the boats in France and the putting down of a re-erecting yard at the port of Varna, including a launching and hauling-up slip, with the necessary tracks, cradles, and mechanical apparatus for laying up the boats during the winter months.

The principal dimensions and main features of the boats are the following: Length, 38 m. (124 feet 8 inches); Breadth, 4.4 m. (14 feet 5 inches); depth of hold, 2.75 m. (9 feet); displacement, about 98 tons; power developed by the engines, 2,000 horsepower; corresponding speed, 26 knots; radius of action at 14 knots, 1,000 miles.

The armament of each boat consists of three 450-millimeter (17.71-inch) torpedo-launching tubes and three 47-millimeter (1.85-inch) quick-firing guns; the ammunition carried being three torpedoes and 390 rounds.

The steam for the engines is supplied by two water-tube boilers, protected by the coal carried in the side bunkers; this affords a protective thickness of 500 millimeters (19.6 inches), equal to an armor plating of 40 millimeters (1.57 inches) thick. As is well known, the attempts at providing torpedo boats with a protective plating have been abandoned, owing to the great increase in the weight of the hull that would ensue thereby, and all efforts made for improving this class of boat have since been towards increasing the speed and armament to a maximum and improving the habitability of the craft.

There are no hard-and-fast rules for designing a coast-defense torpedo boat, and those nations who have recourse to boats of this class for the protection of their coasts have

to take into account, in selecting a type, both their geographical situation and their general naval policy. From the above characteristic features of the Bulgarian boats these boats would appear to form strong craft having a powerful fighting value. Though of light construction, they are strongly built and seaworthy. In service the crew numbers 23 men, including the commander and chief officer.

As is very generally the case, the crew space is in the fore part and the officers' accommodation aft; the necessary measures in regard to coal, provisions and navigability have been taken for facilitating long cruises.

The two du Temple boilers are mounted in the center of the boat, in the same stokehold. Forced draft in closed stokeholds renders intense combustion at high speed possible. The boilers are designed for a pressure of 17 atmospheres (250 pounds per square inch); they have about 220 square meters (2,368 square feet) heating surface. The engine is triple-expansion, working three cranks, and is connected to a surface-condenser. The cylinders are 420, 600 and 900 millimeters (16.53 inches, 23.62 inches and 35.43 inches) in diameter, with a 500-millimeter (19.68-inch) stroke. The engine can easily run at a speed of 330 to 350 revolutions.

An electric generating set supplies current for lighting the boat throughout. An evaporator is provided for renewing the fresh-water supply. The conning tower, in front of the funnels, is equipped with the usual apparatus for the transmission of orders. The boat is fitted with two rudders, which enable turning at high speed round a short circle.

The 47-millimeter (1.85-inch) guns are mounted in sponsons near the conning tower, and have a wide firing field; they can easily aim forward, broadside and abaft the beam. The ammunition holds are quite close, and the guns can quickly be served. One of the torpedo-launching tubes is fitted in the bow; the two others form a double tube, aiming in opposite directions; they are mounted on a platform on deck, near the center of the boat.

The three first boats are afloat; the others are in course of construction at the Varna yard.

The launching of the boats every time they have to carry out a cruise, and their hauling up again, required the putting down of a steam-operated slipway. The site available between the south dyke and the entrance to the Dewna Canal did not allow of putting down the erecting yard opposite the site available for the slipway, neither could the boats be put afloat and hauled up again longitudinally. It was necessary, therefore, to provide for the transversal and longitudinal transport of each unit, from the erecting yard to the slip, and inversely, a difficult problem, seeing that the hulls of the boats are of light construction, and that all the operations have to be effected with the boats fully equipped.

All the elements forming the hulls, galvanized throughout, and delivered to Varna in riveted parts as far as was practicable in regard to transport, were put together on the spot in the ordinary way, on cradles. The stern post and propeller support were bored out when the hulls were fully riveted, with a view to ensure a perfect alignment for the propeller shaft; the engines and boilers were then put in place, and the internal accommodations made at the same time.

Each boat, when thus completed and painted, was launched sideways by means of the slip. This operation comprises two distinct phases—lateral transport in the yard and longitudinal transport to the slip.

Before the first phase, the wood cradle is replaced by a stiff metallic frame carried on rollers and provided with sliding-blocks of the required height. For the lateral transport, the total load, torpedo boat and frame, is coupled to a set of ten jacks, which disengage the sliding blocks and place the boat-carrying frame on the fixed track. The frame and boat are then hauled by means of fixed hand winches. When the three-rail track is reached the frame and boat are lowered by winches on four strong trucks, which are placed previously in the required position and secured by chocks. The transversal traveling gear is then dismounted, and winches fixed at each end of the central track are used to bring the frame and boat on the launching slip. The launching of the cradle and boat is gov-

erned by two steel cables, which are wound on two steam-driven drums. After thus covering a distance of about 27 meters (90 feet) down the slip, the boat commences to float, and frees itself of the cradle.

Hauling up is effected inversely: the boat places itself stem foremost over the cradle, and the uprights on the outside allow the position of the boat to be accurately adjusted.

All the operations are carried out smoothly and without shock. In order to secure a better mechanical efficiency the builders have mounted all movable parts on rollers; they have also given the girders forming the frame a certain degree of flexibility to meet eventual deformations in the various tracks.

The hauling up and dry-docking of a torpedo boat takes about 35 minutes.

The boats are completing their trial trips. The one first built made 27.72 knots on the measured mile, thus exceeding the contract speed by 1.72 nautical miles per hour.—"Engineering."

ENGLAND.

The Gladiator salvage has been a somewhat long and tedious operation, due mainly to the troublesome tides and unpropitious weather. However, the ship is "up" at the moment of writing and should before long now be seen in Portsmouth Harbor. No absolute decision about refitting her seems to have been arrived at. She is a type of vessel now quite obsolete as cruisers go; but for that very reason likely to be useful in a variety of ways. Obsolete ships can be risked where better ones cannot be.

British 33-knotters.—All the first five are now practically completed, and it remains to be seen how naval men will appreciate the... More comfortable and commodious than the older vessels they certainly are, but surely on 850 tons they might carry more than three 12-pounder guns and two 18-inch tubes. In fact, to fit 4-inch guns has already been proposed. Their present armament compares unfavorably with most foreign destroyers now being built, and it certainly seems that

at least two more 12-pounders could have been worked in. The original magazine spaces, however, are now required for oil fuel, and the ammunition supply is perhaps hardly as convenient as it used to be.

Destroyer Building "on spec." has often been attempted by firms specializing in this class of work, not always with the success achieved by Palmers, who had completed two vessels of 400 tons and fitted with turbine machinery, which have just been acquired by the Admiralty to replace the *Gala* and *Tiger*. They are very similar to the *Kangaroo*, but of greater power; and the speeds attained have been well over 31 knots.

H. M. SS. Agamemnon and Indomitable.—The new British battleship *Agamemnon*, built by William Beardmore & Co., Dalmuir, which passed through her speed and consumption trials satisfactorily several months ago, has since undergone "final touches" in the outfitting basin of her builders, and has left the Clyde for Portsmouth, manned by a working crew from the south of England, the results of which are, of course, now well known. But for alterations in the original designs, due to the rapid evolution in naval architecture and, of late especially, to modifications made in the cooling arrangements of the magazines, as the outcome of untoward experiences on board ships of other navies in connection with the deterioration in the quality of cordite, through excessive temperature, this warship, which was launched in June, 1906, a week later than the Cunard turbine *Lusitania*, would have left the Clyde months ago.

The *Agamemnon*, which was laid down in October, 1904, and is a sister ship to the *Lord Nelson*, has a length of 410 feet between perpendiculars, a beam of 79 feet 6 inches, and a mean draught of 27 feet, her displacement being 16,600 tons. Her boilers and machinery were supplied by R. & W. Hawthorn, Leslie & Co., Limited, the designed horsepower being 20,000 and the speed 18½ knots. There are two sets of four-cylinder, vertical, triple-expansion engines. The normal coal capacity is 900 tons, but a maximum of 2,500 tons can be carried, as well as 400 tons of oil fuel. There are

fifteen Babcock & Wilcox boilers. The armament consists of four 12-inch, ten 9.2-inch, fifteen 12-pounders, sixteen 3-pounders, six pompoms and two Maxim guns. There are four submerged 18-inch torpedo tubes on the broadsides, and one submerged tube of similar diameter astern. The armor is of the Krupp cemented type. There is a 12-inch belt amidships and a 4-inch belt at the ends. The barbettes are protected by 14-inch N.C. plate, and the turrets above these by 8-inch K.C. armor, the conning tower having 12-inch N.C. plate to protect it. Two of the 12-inch guns can fire ahead and two astern, the arc of fire being in each case 240 degrees. Four 9.2-inch guns can fire both ahead and astern, and on each broadside four 12-inch and five 9.2-inch can be fired.

The first-class armored cruiser—or as this class has been called the cruiser-battleship—*Indomitable*, launched from the Fairfield stocks last year, after having been docked for hull cleaning in April last, has successfully completed her two-fifths-power machinery trial, and being the first of the three vessels of the *Invincible* class to be completed, is arousing a considerable amount of interest; quite as much, in fact, as was taken in the mighty *Dreadnought*.

The *Indomitable*, as also the other two vessels of her class, the *Inflexible* and *Invincible*, has a displacement of some 17,250 tons. Her length is 530 feet, her beam 78½ feet and her draught 26 feet. She was laid down in March, 1906. Like her sister vessels, she is fitted with turbines of the Parsons type, and with a designed horsepower of 41,000; her contract speed is 25 knots. All the machinery of the three vessels is to be interchangeable throughout. She can carry a maximum quantity of 2,000 tons of coal, but her normal capacity is 1,000 tons. Provision has also been made for carrying oil fuel. Her armament consists of eight 12-inch and sixteen 4-inch guns, and she has three submerged torpedo tubes.

The magnificent run which H. M. S. *Indomitable* has just made has shown that the new battleship-cruiser is capable of speeds hitherto unattainable in any navy, with the ex-

ception of such craft as torpedo-boat destroyers. It is impossible to give precise particulars of a performance concerning which no official information has been supplied. Broadly stated, her passage from Quebec to this country was made in five and a half days, but the number of hours has not been made public, nor the rate at which she came down the St. Lawrence. All that can be said is that leaving Quebec early on Wednesday morning, the 29th ult., the *Indomitable* for the 719 miles down the St. Lawrence River had to steam at a comparatively low speed. At Belle Isle, on the Newfoundland coast, the race home commenced, and the run across the Atlantic was covered at an average speed, variously stated as 25.13 and 24.8 knots. For four hours a speed of 26.4 knots was attained. The great speed at which the cruiser has steamed across the ocean is made all the more remarkable by the fact that this vessel has not been stripped in order to allow her to make a record run. She carried the heavy guns of a *Dreadnought*, a thick coat of armor, her magazines full of ammunition, and all the rest of the equipment of a man of war ready for active service. Her cost was enormous, approaching close upon £2,000,000, and before the year is out another two of these vessels will have been added to the strength of the Navy. The *Indomitable* is 560 feet long on the water line, 78½ feet beam, and has a displacement of 17,250 tons. Her propelling machinery consists of Parsons turbines, capable of developing 41,000 horsepower. She is one of a new group of cruisers, mainly intended for the convoying of fast merchant steamers should war break out. It is stated that on the measured mile she attained a speed of 28 knots. But the official reticence concerning her performance is complete. Steam is supplied entirely by Babcock boilers, made in Glasgow. Probably the most interesting question which she has settled is her steaming range at full power. This may be taken at, in round numbers, 2,500 knots. Her bunkers hold 3,000 tons of coal; how much more she carried, or whether she carried any addition at all, has not been stated. She is, we believe, fitted

to burn oil fuel. We understand that she reached Cowes at 9·40 P. M. on Monday with empty bunkers. Now, it is not likely that her machinery used less than 15 pounds of steam per horse per hour. Even that figure stands for a very fine performance. If her boilers evaporated 10 pounds of water per pound of coal, we have 1½ pounds of coal per horse per hour, or, say, 60,000 pounds, or 27 tons per hour. If her actual steaming time was 132 hours, then she must have burned 3,564 tons on the trip. Two or three hours more or less will not make much difference. Some allowance may perhaps be made for the run down the St. Lawrence. If she had only 3,000 tons of coal to burn, then she must have been helped by oil fuel, or her turbines did not exert 40,000 horsepower. In the latter case the efficiency of the ship as a whole is remarkable. Her propellers must be admirable, and the form of the hull under water reflects the greatest credit on her designer. No doubt, in process of time, the facts of the case will leak out. It is not impossible that they may be published abroad first. From the purely scientific point of view, the performance of the ship is so full of interest, alike to the naval architect and the engineer, that it will be matter for much regret if the publication of the main features of the *Indomitable's* performance is withheld for any considerable time.—"The Engineer."

H. M. S. St. Vincent.—The *St. Vincent*, latest and greatest of the British *Dreadnoughts*, was laid down at Portsmouth on December 30th, and will be launched September 10, 1908. She is the eighth of our all-big-gun ships, including the three vessels of the *Invincible* class. Only the broadest facts about her construction have been made known, and whilst she resembles the prototype in general arrangement, she is larger in every respect and heavier. Her length is 530 feet, her beam 84 feet; she draws 27 feet of water, and displaces 19,250 tons. Her turbines will develop between 24,000 and 25,000 horsepower, and her speed will be about 21 knots. Her armament will resemble closely that of the *Dreadnought*, but the opinion that the 12-pounders of that ship were too

small to repel torpedo attack has been allowed to prevail, and she will carry a number of 4-inch guns. The first of three Brazilian *Dreadnoughts*, which also will be launched September 10th, is the Elswick idea of what an all-big-gun ship should be. Elswick has always favored big armaments, and the *Minas Geraes* carries twelve 12-inch guns to the *St. Vincent's* ten. But it is in the disposition of the armor that we may look in the Brazilian ship for some hint as to the arrangement in the British ship. It must, however, be borne in mind that carrying fewer big guns she is able to bear heavier armor, and in place of the Brazilian 9-inch belt she has a belt amidships 11 inches thick, which probably tapers off considerably fore and aft.

Leading Dimensions of British Battleships.—

.....	*King Edward VII* class.	*Lord Nelson* class.	*Dreadnought* class.
Length B. P.....	425 ft.	410 ft.	490 ft.
Breadth, molded...	78 ft.	79 ft. 6 in.	82 ft.
Draught.....	26 ft. 9 in.	27 ft.	27 ft.
Displacement.....	16,350 tons	16,550 tons	18,600 tons
Speed, designed....	18½ knots	18 knots	21 knots
I.H.P., designed....	18,000	16,750	23,000
Coals, normal.....	950 tons	900 tons	900 tons
Coals, bunkers, fuel	2,150 tons	2,200 tons	2,700 tons
Main armament.....	4 12-in. B. L.; 4 9.2-in. B. L.; 10 6-in. Q. F.	4 12-in. B. L.; 10 9.2-in. B. L.	10 12-in. B. L.
Secondary battery.	14 12-pdr. 12 cwt.; 16 3-pdr.; machine boat and field guns	24 12-pdr. 18 cwt.; 8 3-pdr.; machine boat and field guns.	16 .4-in. Q. F.; machine boat and field guns.
Submerged torpedo tubes.	Four	Five	Three

The British 38-knot Destroyer Swift.—The question of warship speed has been occupying a good deal of attention lately, and general interest in the subject has been considerably quickened by the performance of the United States cruisers of the *Chester* class and of the British *Indomitable.* There have been many assertions made as to what ship is entitled to bear the distinction of being the fastest warship in

the world, and although such discussions have as a rule been confined to vessels of good sea-going and sea-keeping qualities, the question in its wider bearings has been answered very emphatically by the British special-type torpedo-boat destroyer *Swift*. On her preliminary trials this vessel maintained for some hours a speed of 38.3 knots, or nearly 45 miles an hour—higher by three knots than the best four-hour performance ever achieved; and by modifying the propellers it may be possible to get a higher speed out of her.

The *Swift* was laid down in October, 1906, at the works of Messrs. Cammell, Laird & Co. at Birkenhead, and was built to the designs of the builders, modified and improved by Sir Philip Watts, the director of British naval construction. Her displacement is exactly double that of the largest torpedo-boat destroyers previously built, namely, 1,800 tons; while her length of 345 feet falls short by only 36 inches of the length of the 10,300-ton United States battleship *Indiana*. Her beam is 34 feet 2 inches—slightly less than one-tenth of the length—and the mean draught is 10 feet 6 inches.

The *Swift* is, like all recent British ships, fitted with turbine engines on the Parsons principle, designed to develop the stupendous horsepower, for her size, of 30,000, and to give a speed of 36 knots. The turbines are in two compartments, and drive four shafts with one propeller on each. The furnaces are fitted for the burning of oil fuel only, the carrying capacity being 180 tons; and it is the subject of considerable comment that this is no greater than the quantity carried by the 800-ton 33-knot destroyers of the *Tartar* class, which immediately preceded her. The armament of the *Swift* is limited to four 4-inch (25-pounder) rapid-fire guns and two 18-inch torpedo tubes.

After her speed, the most remarkable feature of the *Swift* is her cost. This amounts, in the case of the hull and machinery, to $1,237,310 and to $14,150 for the armament, a total of $1,251,460. This is a huge price to pay for a vessel of only 1,800 tons and practically without any fighting power, and may be profitably compared with the figures given below for

typical cruisers and similar war vessels in the British and in the United States navies.

The greater part of the cost of the *Swift* is, of course, absorbed by her speed; and in this connection it may be interesting to note that if the *Indomitable* had been designed for 23 knots instead of 25, it is estimated that she would have cost $1,500,000 less than she actually did, and that if the *Dreadnought* had been designed for 18.5 knots instead of 21, she would have cost $2,150,000 less. Since Great Britain has four *Indomitables* and eight *Dreadnoughts* built, building or projected, the total saving would have been no less than $24,000,000—sufficient to build another three battleships.

It is not known whether the British Admiralty intend to repeat the *Swift*, but it is regarded as very improbable. At a time when it is so difficult to get money from the Government for purposes of national defense, it is likely that the Admiralty will find some more substantial way of spending money than in the creation of speeds which, however startling, have but a very limited military value.

Ships.	Type.	Displacement normal, tons.	Speed, nominal, knots.	Armament.	Cost.
Swift, (British).....	Destroyer	1,800	36	Four 4-in.	$1,251,460
Adventurer, (British).....	Scout	2,940	25	Ten 3-in.	1,142,130
Amethyst, (British).....	Cruiser.	3,000	23	Twelve 4-in.	1,142,130
Chester, (United States)..	Scout	3,750	26	{ Two 5-in. { Six 3-in.	1,625,000*

* Exclusive of armament.

—"Scientific American."

FRANCE.

The French have lately been fairly busy demolishing some of their old ironclads. The old monitor *Tempête* has just been fired at by the Mediterranean Fleet. Exact details of the firing are not obtainable, but substantially the results

were as follows: (1) The superstructure was blown entirely to pieces. (2) The mast was grazed by a number of projectiles which eventually brought it down. (3) The port side was fired at and the unarmored parts absolutely "honeycombed—with hits" a series of neat little holes. The starboard side, the side away from the hits, was badly damaged by shells bursting after getting into the ship. This side seems to have burst most of the shells. (4) The armor was several times hit, but in no case penetrated. (5) The range was considerable—8,000 yards according to some accounts. (6) The captains of guns fired first and secured 16 hits out of 40 rounds. Their successors got an average of 13 hits out of 80 rounds.

The old battleship *Admiral Duperré* was fired at and sunk by the Toulon forts, but details were kept confidential, as special projectiles were employed.

GERMANY.

The New German Dreadnoughts.—The recent Admiralty return giving the armament of the new German *Dreadnoughts* led to much discussion about the way the details had been kept secret. As a matter of fact, their principal features had long been known in those circles which were interested in ascertaining them, together with much more which detracts in naval eyes from the paper attractions of twelve 11-inch and twelve 6.7-inch guns. At the ranges at which ships of the *Dreadnought* class were designed to fight, the energy in an 11-inch shot is relatively very much less than in the case of a 12-inch compared with the energy at the muzzle. At the ranges and for the purposes for which the secondary armament is considered to be required in the case of the *Dreadnought*, the greater rapidity of the fire from the 4-inch is much preferred to the slower 6.7-inch. Moreover, the *Nassau*, with her very congested engine rooms and triple screws, will have to exceed her designed speed if she is not to be at least 1½ knots slower than the *Dreadnought*. Still more, the *Temeraire* class are improved *Dreadnoughts*, while the *Nassaus*

are, strictly speaking, cotemporary with the *Dreadnought* herself.

According to "Le Yacht," the armaments of the German *Dreadnoughts* are as follows: *Nassau* type, twelve 11-inch, twelve 4.7-inch; *Ersatz Baden* type, fourteen 11-inch, twelve 6.7-inch or fourteen 4.7-inch; *Ersatz Beowulf*, twelve 12-inch, twelve 6.7-inch; and the displacements 18,000, 19,000 and 20,000 tons respectively.

The same authority gives the cruisers as follows: "*E*" (*Blucher*), ten 11-inch; "*F*," twelve 11-inch and some 4.7-inch; "*G*," ten 12-inch; the displacements being 15,000, 19,000 and 22,000 tons.

The orders for the twelve new German destroyers of the 1907–8 program were recently given out. As at present arranged they are to be of 600 tons displacement and about 13,000 horsepower. A speed of 30 knots is to be guaranteed, but more is hoped for in view of the performance of *G 137*. All will be turbine driven—four by Parsons turbines, four by Melms and Pfenniger, three by Curtis and one by Zoelly turbines.

ITALY.

New Destroyers.—The Ansaldo Company has laid down no less than six stock destroyers of about 350 tons and 6,000 indicated horsepower. The Italian navy has been building a large number of boats of this type recently, carrying four 12-pounder guns and three tubes, but is now reported to be considering boats of the ocean-going type, with speeds of 30 knots.

RUSSIA.

Trials of Armored Cruiser Rurik.—The *Rurik*, designated an armored cruiser, but in some respects more closely resembling a battleship, built by Messrs. Vickers Sons & Maxim, Limited, at their Naval Construction Works, Barrow-in-Furness, has concluded, with marked success, probably the most exhaustive series of trials yet specified for any warship, and will shortly leave this country for Russia, to be added to the

navy of his Imperial Majesty the Czar. The vessel, which embodies not only the great experience of the Vickers Company, but also the valuable suggestion of the Russian Admiralty as an outcome of the recent war, is in many respects as notable a departure from former practice in naval design as was the *Dreadnought*. Indeed, she is the antithesis of what has been termed "the paper ship." In the first place the *Rurik* has a very powerful armament, not only for service in the line of battle, but also for repelling torpedo-boat attack. All of the guns have a greater angle of elevation and depression than in previous ships, as well as a large arc of training, and all are operated by electrical mechanism. In the second place, the vessel is specially notable for the effective character of her defence, as she is armored practically from the keel to the upper deck. In the third place, she has a great reserve of boiler power, the heating surface provided being equal to 2.86 square feet per indicated horsepower at full speed, and, as will presently be explained, she was required to maintain full speed for three hours with only three-quarters of this boiler power, so that the full speed of 22 knots is more comparable with the speed attained, even in British practice, with only 75 per cent. of the boiler power. In the fourth place, the vessel carries a greater supply of ammunition than is usually provided, and was, by the wish of the Russian Commission, loaded down on her speed trials to a displacement exceeding by 300 tons the correct displacement. As a consequence of these circumstances, any comparison of the speed performance of the *Rurik* with the speed of other vessels less effectively designed for attack or defence must necessarily be qualified, regard being had to the general scheme to make the vessel as reliable as possible in all conditions of warfare. This view has animated the builders and the Russian Naval Commission, composed of between twenty and thirty officers of great theoretical and practical ability and experience. In testing every detail of mechanism in the ship, it was a condition that wherever even the slightest adjustment was necessary, the whole trial should be repeated after the adjustment was

made. It was not enough that all concerned might be satisfied that the result must be satisfactory on the principle that effects follow inevitably upon incontrovertible conditions. Everything was done rather to thoroughly establish reliability by proved results, and the *Rurik* is therefore a most important addition to the navy of the Czar.

The designed displacement of the ship was 15,200 tons on a draught of 26 feet, and the realized condition is well within these limits. The length between perpendiculars is 490 feet and the molded breadth 75 feet.

The *Rurik* has four 10-inch breech-loading guns, 50 calibers in length, twin-mounted in barbettes forward and aft on the center line. Each of these guns can be worked through 35 degrees of elevation and 5 degrees of depression, and the forward guns can be trained 45 degrees abaft the beam, and the after guns 45 degrees before the beam. There are also eight 8-inch breech-loading guns, also 50 calibers in length, twin mounted in barbettes on the quarters of the ship, with a correspondingly large range of fire to that of the larger guns. For repelling torpedo attack there are twenty 4.7-inch quick-firing guns, also of 50 calibers in length ; sixteen of these are placed within an armored battery in the center of the ship, separated from each other by traverses of specially-hardened armor. This battery, being on the upper deck, enables the secondary armament to be placed higher above the water line than usual ; it also incidentally adds very largely to the armored protection, as it is above, and additional to, the normal armored side of similar ships. Aft there are four 4.7-inch guns, and these are also within armor on the sides of the ship, while to counteract the effects of raking fire three armored bulkheads have been fitted ; this is quite a new feature in the protection of warships. There are twelve smaller quick-firing guns. Two 18-inch torpedo tubes, completely submerged, are placed forward. Each of the guns in the ship is capable of developing a much higher muzzle energy than weapons of the same caliber hitherto fitted, so that from all points of view the armament of the ship is very formidable.

The exhaustive character of the ordnance trials is indicated by the fact that one hundred rounds—full charges—were fired from one of each type of gun, these guns being similar to the ship's guns, and built specially for trial purposes. Following this, thirty rounds were fired from two of the 10-inch guns and two of the 8-inch guns, and fifteen from the other guns at various angles of elevation and depression, and on various bearings, in order to test the mountings of the respective guns. The separate items of the mechanism had formerly been subjected to many trials in order to ensure reliability, especially in view of the departures from earlier prac tice, due to the extensive application of electricity. In this the Russian naval authorities have shown commendable progress. The care bestowed on adjustment of the mountings resulted in the trials at sea being successful. The respective gun mountings were worked to give, when firing ten rounds, a rapidity of two rounds per minute from the 10-inch guns, of three rounds from the 8-inch guns, and of eight rounds from the 4.7-inch guns, while the 47-millimeter guns fired at the rate of between twenty and thirty rounds, and the Maxim gun at about five hundred rounds per minute.

In regard to the defensive qualities of the design, the vessel has a complete belt from end to end, extending to a considerable distance below the water line. It is, for the greater part of the length, 6 inches in thickness, tapering to 4 inches and 3 inches at the extreme ends of the hull. For protecting the 4.7-inch guns, the upper strake, for 200 feet of the length of the ship, is of 3-inch specially-hardened armor, manufactured, like all the other armor, at the Sheffield works of the Vickers Company, and subjected to careful tests at the Eskmeals range. The barbettes, which are within this central battery, are of much heavier armor, having 7½-inch walls, while the conning towers are of 8-inch armor. Another feature is the range-finding towers, of which there are two, extending for a considerable height above the upper deck, and constructed of 5-inch specially-hardened armor. These offer, independently of the mast, satisfactory observation stations for the determination of the range for gun fire.

There are protective decks of a combined thickness of 4 inches, arranged so as to secure as far as practicable that all high-explosive shells shall burst outside of the ship. The base of each of the three funnels is protected by armored casings. The whole of the machinery and magazines which are under the water line are surrounded by armored walls, which extend from the main deck, through the protective deck, to the bottom of the ship. This is in addition to the usual construction of double skin. Thus all the machinery and vital parts of the vessel are adequately protected by specially-hardened armor.

The machinery, as we have already stated, has been designed with less regard to economy of weight than to reliability under any adverse conditions. The boiler power is considerably in excess of that provided in former practice. There are twenty-eight water-tube boilers, and these are fitted for consuming coal or oil fuel. The propelling machinery consists of two sets of four-cylinder triple-expansion engines, carefully balanced to obviate vibration and to make the ship a steady gun platform. Each set of engines has one high, one intermediate, and two low-pressure cylinders, the power under easy steaming being 20,000 indicated horsepower, with the boilers working at a steam pressure of 285 pounds to the square inch, which is reduced to 250 pounds before entering the engines. A complete system of auxiliary machinery has been fitted, and special attention has been devoted to the pumping and drainage arrangements, which are the outcome of much thought on the part of the Russian Admiralty officials.

The steam trials were conducted under the supervision of a large Commission of experienced Russian officers, who, as we have already indicated, have been actuated with the one aim of ensuring that when the vessel goes into commission she shall be thoroughly efficient even in the hands of officers without experience of this particular ship. As a consequence, the trials have been more protracted than is usual in other services where the best reliable results are got after

what is known in engineering language as the grinding down of bearings during the first few months of commission. In other words, the *Rurik*, as a result of her trials, is now in the position of other vessels which have been several months in commission. Each of the requirements of the specification prepared by the Russian Admiralty has been fully satisfied, and a study of the official results which we are enabled to publish shows that this ship, notwithstanding the very important innovations to ensure increased strength in attack and defence, is capable of an exceptionally high speed in continuous steaming, and even with some of the boiler power out of action. Perhaps the most interesting features of these trials were those at full power. In the British service the maximum power of the machinery has to be maintained for eight hours. In the case of the *Rurik* this condition had to be kept up for ten hours, and, in addition, as we have already indicated, the vessel had to run for three hours at 21 knots with one-fourth of the boiler power entirely out of service.

Moreover, during the usual twenty-four-hours' completion trial, made immediately before the acceptance of the ship and after the opening out of the machinery, it was decided to run the machinery for ten hours at full power. In the British service this commissioning twenty-four-hours' trial very rarely includes more than one-hour's run at full power. The *Rurik* on this trial made a splendid performance, as the speed maintained for the ten hours was much more than that required by the contract. The results are given in column 6 of the annexed table. The mean indicated horsepower was 20,675, as compared with 19,700 horsepower guaranteed; the mean revolutions being 141.6, against the 135 required for 21 knots. It will thus be recognized that the revolutions, power and speed are in excess, but in correct proportion, indicating the accuracy of the design and the realization of the usual marginal allowance. Moreover, the engines did not vary more than one revolution per minute, and, with the exception of one reading, the hourly results ranged between 20,000 and 21,000 horsepower. The earlier ten-hours' trial, of which the

TABLE SHOWING RESULTS OF TRIALS OF THE *RURIK*.

1	2	3	4	5	6
.....	Mean of 30 hours at 12 knots.	Mean of 30 hours at 19 knots.	Mean of 10 hours at 21 knots.	Mean of 3 hours at 21 knots with three-quarter boiler power.	Mean of 10 hours at full power during 24-hours' completion trial.
Steam pressure, pounds per sq. inch....	196.0	261.0	269.0	273.0	280.0
Vacuum, inches.....	27.12	27.0	26 0	25.8	26.2
Revolutions per minute.....	75.6	123.85	138.48	138.3	141.6
Indicated horsepower.....	3,039	13,359	19,355	18,953	20,675
Number of boilers in use.....	8	21	28	21	28
Air pressure in stokeholds, inch.....	0.13	0.3	0.31	0.58	0.36

results are given in column 4 of the table, was equally satisfactory, and the three-hours' trial with three-fourths of the boiler power in use, the leading points of which are given in column 5 of the table of results, was still more remarkable.

The speed was determined during the trials as a consequence of several runs made over the measured mile at Skelmorlie, on the Clyde. The condition stipulated was that throughout the ten hours, and later throughout the three hours, the engines should keep up an average number of revolutions equaling or exceeding the number corresponding to 21 knots, as ascertained on the measured-mile trials, and this was easily accomplished. The *Rurik* also carried through two thirty-hour trials, one at about 19 knots and the other at about 12 knots. These trials were to determine the radius of action, and the results are given in the table.

Very careful data were collected by the Commission, not only as to coal consumption, but also as to water consumption, tanks being specially provided. An important condition in the contract had reference to stability, and the specified metacentric height was exceeded by about 10 per cent., with which result the Russian officers were highly pleased. Observation was also taken of the temperatures in various parts of the ship, &c. Maneuvering trials were also

carried out, and in this respect the ship was found to be very satisfactory. Indeed, from the first to the last, the Russian Commission put the ship through a series of tests of a most searching character, and the result establishes that the severe conditions of the contract have been fulfilled, and that the *Rurik* is from every standpoint a remarkable ship of the line.—"Engineering."

SPAIN.

The New Navy.—Tenders have already been submitted for the new Spanish Navy. The battleships are to be of about 15,000 tons, 19½ knots speed and about 425 feet long. Three destroyers of 360 tons and 28 knots, and twenty-four torpedo boats of 180 tons and 26 knots, as well as four small gunboats of about 1,000 tons, complete the proposed fleet. The large ships are to be built at Ferrol and the smaller vessels at Carthagena. Both dockyards are to be entirely renovated, and it is estimated that the work will occupy at least eight years.

OBITUARY.

CHARLES WHITESIDE RAE, REAR ADMIRAL, U. S. N., Chief of the Bureau of Steam Engineering.

Rear Admiral Charles Whiteside Rae, Engineer-in-Chief of the Navy and Chief of the Bureau of Steam Engineering, Navy Department, died at his home in Washington, D. C., the evening of May 13, 1908.

Rear Admiral Rae was born at Hartford, Ct., June 30, 1847. After a preliminary course of study at the Champlain Academy, New York, he entered Rensselaer Polytechnic Institute and was graduated with the Class of 1866. Being attracted towards a Naval life, he entered the Navy as an Acting Assistant Engineer on October 10, 1866. Together with a number of other Acting Assistant Engineers he was sent to the Naval Academy for a two years' course of instruction, and was graduated from that institution in 1868.

After some period of sea service we find him in charge of the work of establishing a line of levels across the Isthmus of Tehuantepec (1870), and shortly after that he was attached to the Nicaragua Canal Survey Expedition. After this he was detailed to the Naval Academy as an instructor for the class of Cadet Engineers, remaining on that duty for several years. He saw duty on the *Lancaster*, European and South Atlantic Stations, and on the *Atlanta*, of the North Atlantic Squadron.

Admiral Rae was detailed to the important duty of head of the Department of Steam Engineering at the Naval Academy in September, 1893, and remained on that duty until June, 1897. During the Spanish war he was Chief Engineer of the Battleship *Iowa*, and participated in the bombardment of San Juan, Porto Rico, in the battle of Santiago, and in

several minor actions off the coast of Cuba. For "eminent and conspicuous conduct" during the battle of Santiago he was advanced three numbers, and for distinguished service in the Cuban campaign he was awarded a medal of honor.

For about a year after the Spanish war Admiral Rae had charge of the installation of new machinery for Yerba Buena Naval Station, and was then detailed as Inspector of Machinery for the Navy at the works of the Newport News Shipbuilding and Dry Dock Co., leaving this latter duty for duty under the Bureau of Steam Engineering in Washington. On August 9, 1903, President Roosevelt appointed him Engineer-in-Chief of the Navy and Chief of the Bureau of Steam Engineering, with the rank of Rear Admiral, which position he filled up to the time of his death, having been reappointed by the President in 1907 for a second term of four years. In 1890 he married Miss Rebecca Gilman Dodge, of Washington, D. C., who survives him.

Rear Admiral Rae was a Vice President of the Society of Naval Architects and Marine Engineers, a past President of the American Society of Naval Engineers, a past President of the Washington Society of Engineers, a member of the National Geographic Society, and a member of several prominent clubs in New York City and in Washington. In 1906 he received the degree of Doctor of Science from the University of Pennsylvania.

In the death of Rear Admiral Rae the Navy has lost a capable and efficient officer; one who worked quietly and well, and whose influence will long be felt for good in the field of his chosen profession. By precept and by example Admiral Rae strove to place Naval Engineering on the high plane that its importance deserves. His own rise from the junior grades of engineering duty to the head of the Naval Bureau of Steam Engineering was a fitting reward for duty well done. Admiral Rae was a man of charming personality. Courteous and dignified in bearing, he attracted all with whom he came in contact and made friends even of those whose pet schemes he could not favor.

BOOKS RECEIVED.

HANDBOOK FOR THE CARE AND OPERATION OF NAVAL MACHINERY. By LIEUTENANT H. C. DINGER, U. S. Navy. This excellent work consists of 295 pages, and is made up principally from a series of articles that appeared in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS by this author.

It will serve to fill a demand for a concise and simple description for the care and operation of naval machinery on many points not largely treated of in standard works on marine engineering.

The information contained in the Handbook has been derived from the personal experience of the author, and from consultation with Engineer Officers in the Naval Service.

It consists of thirty-two chapters, with one hundred and twenty-four illustrations.

The book is divided into six parts, as follows:

Part 1. Operation of Naval Machinery;
2. Care and Overhaul of Main Plant;
3. Fittings and Auxiliaries;
4. Care and Preservation, Subdivision of Hull;
5. Special Auxiliary Engines;
6. Spare Parts and Tests.

The publishers are the D. Van Nostrand Company, 23 Murray and 27 Warren Streets, New York. The price is $2.00.

HYDRAULICS. By S. DUNKERLEY, Professor of Civil and Mechanical Engineering in the University of Manchester and Director of the Whitworth Laboratories. Volume II, treating of the Resistance and Propulsion of Ships, of this

work has been received. This work treats in an exhaustive manner the subject of resistance and the propulsion of ships.

It is divided into six chapters, as follows:

Chapter 1. Stream Lines;
2. Waves;
3. Resistance of Ships: Eddy, Skin and Wave-making Resistance;
4. Wave-making Resistance;
5. Trials on Full-sized Ships;
6. Theoretical Considerations Affecting the Propulsion of Ships.

The book is published by Longmans, Green and Company, of London, New York, Bombay and Calcutta. The price is $3.00 per volume.

ASSOCIATION NOTES.

At a special meeting of the Council, held August 31, 1908, Commander W. W. WHITE, U. S. N., Retired, was elected a Member of the Council to fill the vacancy caused by the transfer of Commander B. C. BRYAN, U. S. N., to the Navy Yard, Philadelphia, Pa.